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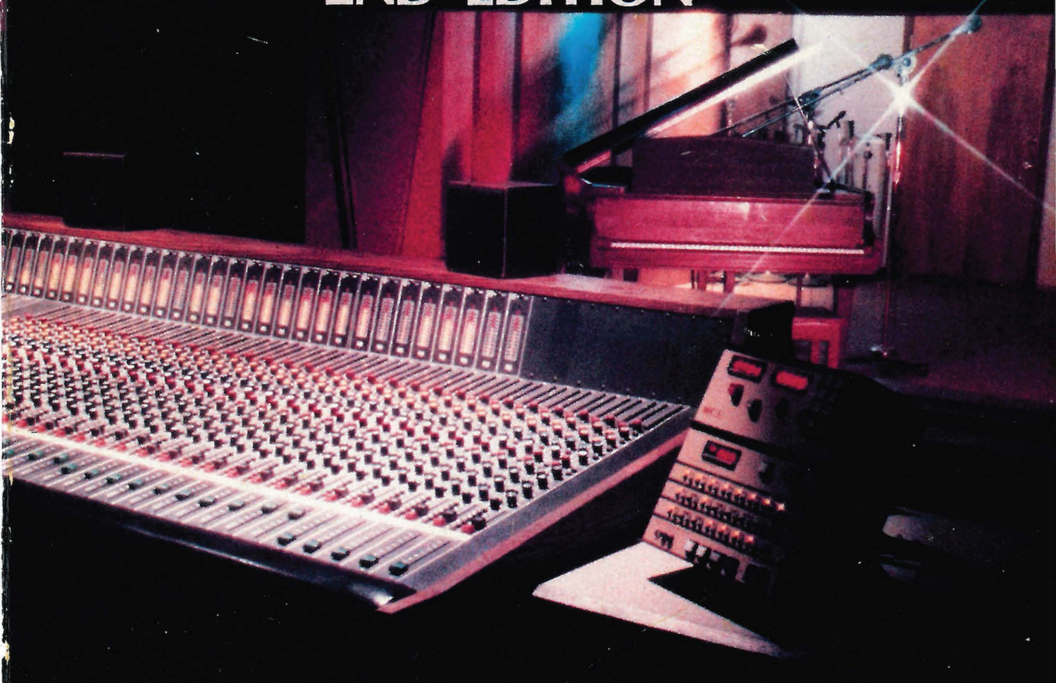
ACOUSTIC TECHNIQUES FOR HOME & STUDIO

2ND EDITION



An essential sourcebook
for every dedicated audio enthusiast!

BY F. ALTON EVEREST

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FOR HOME & STUDIO
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Introduction

It is a continual source of amazement to see how much interesting and useful information lies buried in technical tomes—specialist talking to specialist in languages only a select few can understand. This book explains some of the important acoustical concepts for the audio practitioner, hi-fi enthusiast, and professional alike. The latest findings of the research laboratories are applied to the acoustical design of listening rooms (both monitoring rooms and home installations) and recording studios. The principles elucidated apply to the basement/garage/living room audio setup as well as the professional studios and control rooms.

The first edition of this book appeared in 1973. During the intervening decade, acoustical knowledge has exploded, recording facilities have proliferated, and the typical hi-fi installation boasts equipment with excellent performance characteristics. In fact, the quality of audio equipment has far outstripped the quality of the acoustical environment in which it functions in both the professional and non-professional field. This new edition retains the original goal of speaking in simple, understandable terms and, in greatly expanded form, applies the latest findings of acoustical research laboratories to current audio problems.

New chapters on the decibel and listening room design have been added. Original chapters on fundamentals of sound, room resonances, human hearing, and the acoustical design and tuning of audio rooms have been greatly expanded. References in this edition will be found at the end of each chapter. The pioneering nature of the first edition still prevails in this greatly expanded revision.

1

How Sound Acts

"If a tree falls in a forest with no ear to hear it, is any sound produced?" This ancient question is as modern as tomorrow because it brings us face to face with the dual nature of sound. Sound may be defined as a wave motion in air or other elastic medium (stimulus), or as that excitation of the hearing mechanism which results in the perception of sound (sensation). Which definition applies in a given situation depends upon whether our approach is physical or psychophysical. Viewed in this way the ancient question can be readily answered. There would certainly be the physical stimulus as the tree falls, but if no ear is present there could be no sensation. The type of problem we are facing would thus dictate our approach to sound. If we are interested in the disturbance in air created by a loudspeaker, we approach it as a problem in physics. If we are interested in how it sounds to a person nearby we must involve psychophysical methods. In this book we are concerned with acoustics in relation to people and must therefore treat both aspects of sound[1].

We can break these two views of sound down into terms that are more a part of the experience of the audio enthusiasts:

Physical Quantity	Comparable Psychophysical Quantity
Frequency	Pitch
Intensity	Loudness
Waveform (Spectrum)	Quality (Timbre)

Frequency is a nice substantial characteristic of periodic waves with which we are quite familiar. We measure frequency in hertz (cycles per second); we can observe frequency on a cathode ray oscilloscope, and 100 Hz has a strong tendency to remain 100 Hz. When our ear perceives it, however, the *pitch* for a soft 100 Hz tone may be quite different than for a loud one. The pitch of a low frequency tone goes down and that of a high frequency tone goes up as intensity is increased. A famous acoustician, the late Dr. Harvey Fletcher, found that playing pure tones of 168 and 318 Hz at modest level produces a very discordant sound. At a high intensity, however, the ear hears them in the 150-300 Hz octave relationship as a pleasant sound. For such reasons we cannot equate frequency and pitch, we can only say that they are analogous.

The same situation exists between *intensity* and *loudness*, as the relationship between the two is anything but linear. This will be studied in considerable detail in Chapter 2 because of its great importance in high fidelity work.

Similarly, trying to relate measured *waveform* (or spectrum) and perceived *quality* (or timbre) is complicated by the functioning of the hearing mechanism. As a complex waveform may be described in terms of a fundamental and a train of harmonics of various amplitudes and phases (as we shall see in more detail later), the frequency-pitch interaction would be involved as well as other factors.

In the remainder of this chapter we will deal with the physics of sound in very elementary terms, concentrating on those phenomena of special interest in listening rooms and studios[2]. Chapter 4 treats the ear, how it functions, and something of its non-flat, non-linear characteristics which have to do with our perception and judgment of quality of sounds. Chapter 3 considers the signals with which we have to deal, both wanted and unwanted. With a reasonable understanding of these three chapters, the problems of high fidelity may be viewed in a new and helpful perspective.

HOW IS SOUND TRANSMITTED?

Imagine, in the laboratory, an electrical buzzer suspended within a heavy glass bell jar. As we push the button we hear the buzzer through the glass. As the air is pumped out of the bell jar, however, the sound becomes fainter and fainter until it is no longer audible. The sound conducting medium, air, has been removed from between the source and the ear. Sound must have a medium or it cannot be transmitted from one point to another. Air is a very

common agent for the conduction of sound although other gases as well as solids and liquids conduct sound also. As outer space is an almost perfect vacuum, no sound can be transmitted except in the tiny island of air (oxygen) within a spaceship or a spacesuit.

In Fig. 1-1 we have a weight W suspended from a spring. If the weight is pulled down to the −5 mark and released, the spring will pull it back toward zero. However, the weight will not stop at zero; its inertia will carry it on past zero almost to +5. The weight will

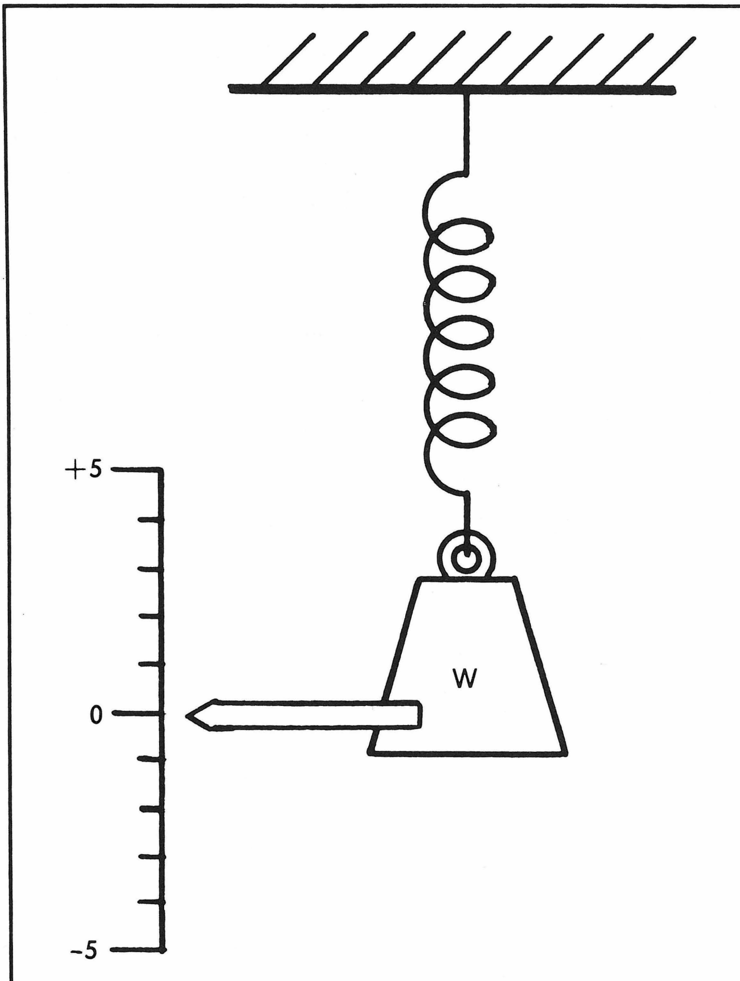


Fig. 1-1. A weight on a spring vibrates at its natural frequency because of the elasticity of the spring and the inertia of the weight.

continue to oscillate or vibrate at an amplitude that will slowly decrease due to frictional losses within the metal of the spring, in the air, etc.

In the arrangement of Fig. 1-1, vibration or oscillation is possible because of two properties, the *elasticity* of the spring and the *inertia* of the weight. Elasticity and inertia are two things all media must possess to be capable of conducting sound. If an air particle is displaced from its original position, elastic forces of the air tend to restore it to its original position. Because of the inertia of the particle, it overshoots the original position, bringing into play elastic forces in the opposite direction, and so on.

Sound is readily conducted in gases, liquids, or solids such as air, water, steel, concrete, etc., which are all elastic media. As a child, perhaps, you have heard two sounds of a rock hitting a railroad rail in the distance, one sound coming through the air and one through the rail. The one through the rail arrived first because the speed of sound in the dense steel is greater than that in tenuous air. Sound also travels well in the sea at a speed about five times that in air. The SOFAR system has given pinpoint locations at sea by sounds traveling over a thousand miles through the water. The cacophony of sounds in the sea has been revealed to be the purposeful, communicating sounds made by whales, porpoises, seals, fish of many kinds, and other creatures.

THE DANCE OF THE PARTICLES

Waves created by the wind travel across a field of grain yet the individual stalks remain firmly rooted as the wave travels by. Neither do particles of air propagating a sound wave move far from their undisplaced positions as shown in Fig. 1-2. The disturbance travels on but the particles do not. The particles of the medium do their little dance close to home.

There are three "dances of the particles". If a stone is dropped onto a calm water surface, concentric waves travel out from the point of impact, the water particles describing circular orbits (for deep water at least) as in Fig. 1-3(A). Another kind of wave motion is illustrated by a violin string (Fig. 1-3(B)). Here the tiny elements of the string move transversely or at right angles to the direction of travel of the waves along the string. We are primarily interested in sound traveling in a medium such as air. In this case, Fig. 1-3(C), the particle vibrates back and forth in the direction the sound wave is traveling. These are called *longitudinal waves.*

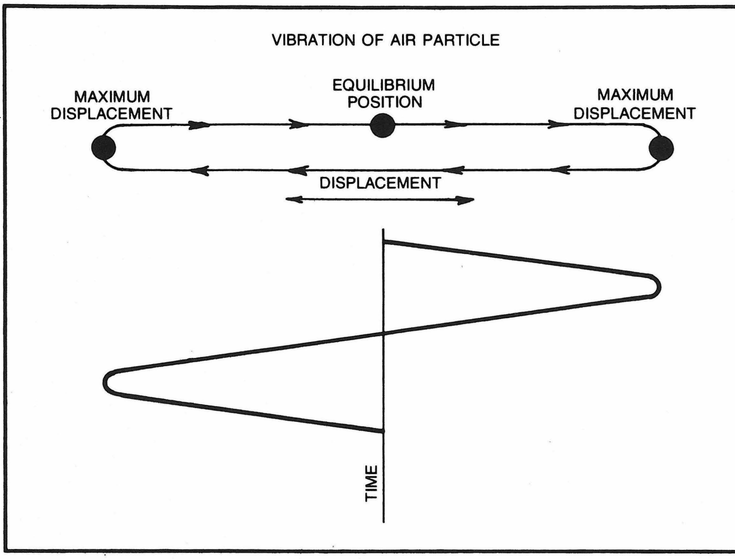


Fig. 1-2. An air particle is made to vibrate about its equilibrium position by the energy of a passing sound wave because of the interaction of the elastic forces of the air and the inertia of the air particle.

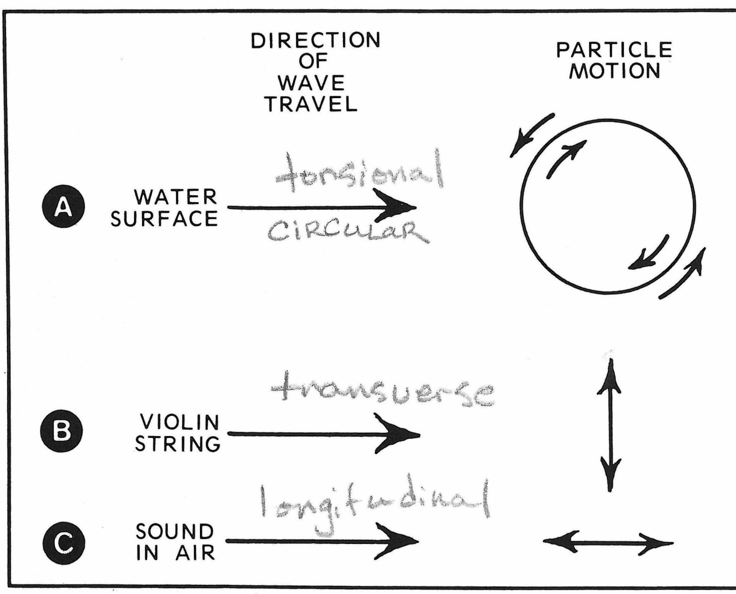


Fig. 1-3. Particles involved in the propagation of sound waves can dance in circular, transverse, or longitudinal motions.

HOW A SOUND WAVE MOVES

Now, how are air particles jiggling back and forth able to carry that beautiful music from the loudspeaker to our ears at the speed of a rifle bullet? In Fig. 1-4 the little dots represent air molecules. This sketch is exaggerated, as there are over a million, million, million molecules of air in a cubic inch. Figure 1-4(A) represents a sound wave traveling from left to right and Fig. 1-4(B) the same thing an instant later. The molecules crowded together represent areas of *compression* (pressure higher than atmospheric) and the sparse areas represent *rarefactions* (pressure lower than atmospheric). The small arrows tell us that on the average the molecules are moving to the right on the compression crests and to the left in the rarefaction troughs between the crests. Any given molecule will move a certain distance to the right and then the same distance to the left of its undisplaced position as the sound wave progresses uniformly to the right.

What makes the sound wave move to the right? The answer is revealed by a closer look at the small arrows of Fig. 1-4. The

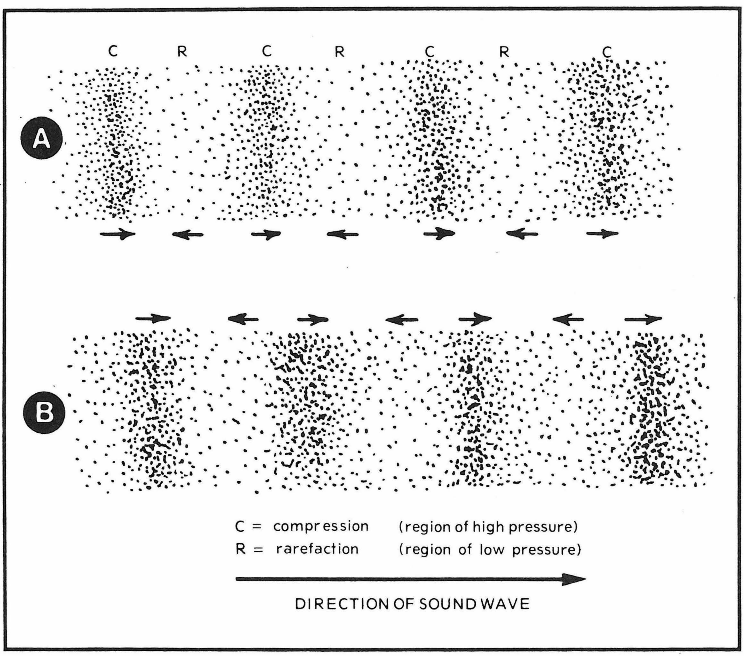


Fig. 1-4. In (A) the wave causes the air molecules to be pressed together in some regions and spread out in others. An instant later (B) the wave has moved slightly to the right.

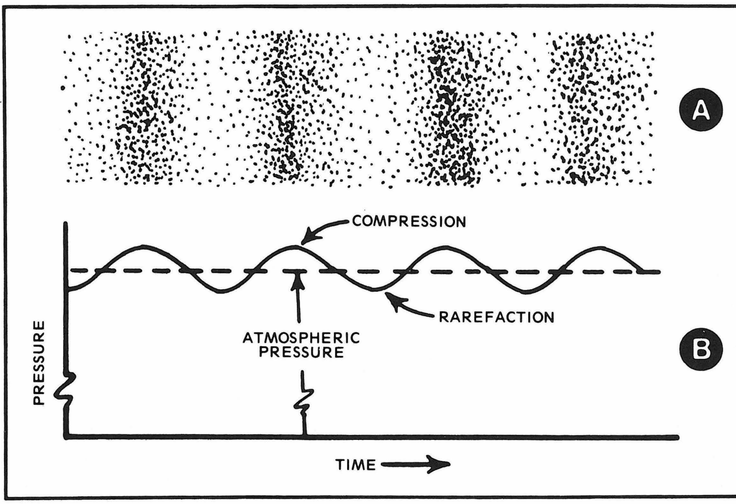


Fig. 1-5. (A) An instantaneous view of the compressed and rarefied regions of a sound wave in air. (B) The compressed regions are very slightly above and the rarefied regions very slightly below atmospheric pressure. Pressure variations of sound waves are thus superimposed on prevailing barometric pressure.

molecules tend to bunch up where two arrows are pointing toward each other and this occurs a bit to the right of each compression. When the arrows point away from each other the density of molecules will decrease. Thus the movement of the higher pressure crest and the lower pressure trough account for the small progression of the wave to the right.

At the crests, the pressure is higher than the prevailing atmospheric pressure as read on the barometer; in the troughs, lower (see Fig. 1-5). The fluctuations (above and below the atmospheric pressure) that represent the sound wave may be very small indeed. For example, the faintest sound the human ear can hear (20 μPascal) is some 5,000 million times smaller than atmospheric pressure.

THE SINE WAVE

The weight of Fig. 1-1 moves in what is called simple harmonic motion. If a ballpoint pen is fastened to the pointer and a strip of paper is moved past it at a uniform speed, the resulting trace is a sine wave (Fig. 1-6). The air particles of Fig. 1-5(A) are set in motion by the tines of a tuning fork which also move in simple harmonic motion and which also create a sine wave pattern of compressions and rarefactions of the air particles, which can be represented by the graph of Fig. 1-5(B). The upward loops repre-

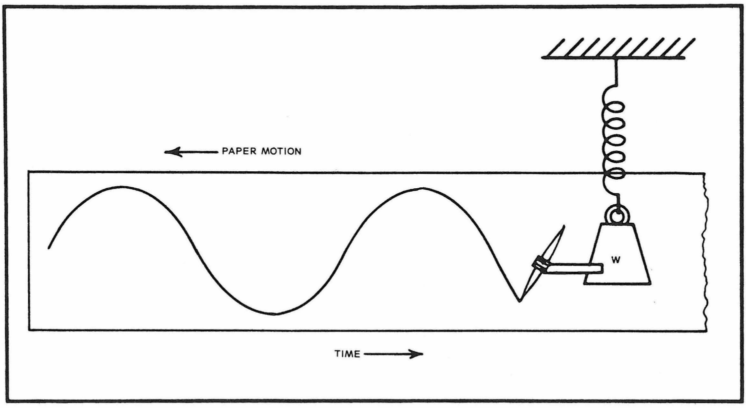


Fig. 1-6. A ballpoint pen fastened to the weight vibrating up and down traces a sine wave on a paper strip moving at uniform speed. This shows the basic relationship between simple harmonic motion and the sine wave.

sent the compressions and the downward troughs the rarefactions, which vary above and below the average static atmospheric pressure prevailing at the moment.

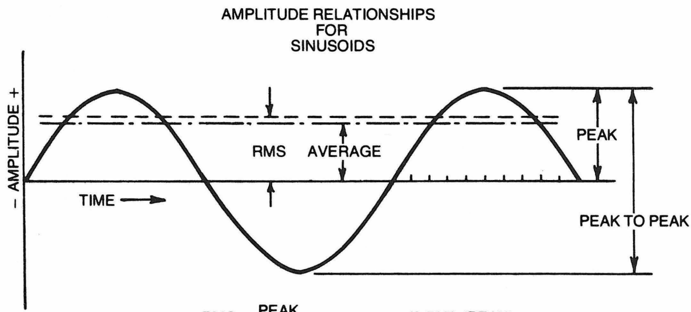
The shape of this graph, the sine wave, is a very special form that has great significance in mathematics, electronics, and acoustics. It is the simple form, the elemental building block of which all complex waves are constituted, as we shall see in Chapter 3.

SINE WAVE LANGUAGE

A very logical question would be, "How can any definite value be given an alternating sine wave of electrical current or voltage or sound pressure when its value is changing all the time from a positive peak to zero and on to a negative peak and then through the whole cycle all over again?" Let's hold that question for a moment and we shall see that the answer lies in what is called its effective value.

Whether in electronics, acoustics, or mechanics, the various terms associated with alternating signals in general and the sine wave in particular permeate the literature, hence it is well to become familiar with them. Viewed on an oscilloscope, the easiest value to read off the scale is the peak value (of voltage, current, sound pressure, or whatever the sine wave represents), the meaning of which is obvious in Fig. 1-7. If the wave is symmetrical, the peak-to-peak value is twice the peak value.

The common ac voltmeter is, in reality, a dc instrument fitted with a rectifier which changes the alternating sine current to pul-



$$\text{RMS} = \frac{\text{PEAK}}{\sqrt{2}} = (0.707)\,(\text{PEAK})$$

$$= \frac{\pi}{2\sqrt{2}}\,(\text{AVERAGE}) = (1.1)\,(\text{AVERAGE})$$

$$\text{PEAK} = (\sqrt{2})\,(\text{RMS}) = (1.414)\,(\text{RMS})$$

$$= \left(\frac{\pi}{2}\right)(\text{AVERAGE}) = (1.57)\,(\text{AVERAGE})$$

Fig. 1-7. Amplitude relationships for sinusoids which apply to sine waves of electrical voltage or current as well as to acoustical parameters such as sound pressure.

sating, unidirectional current. The dc meter then responds to the average value as indicated in Fig. 1-7. However, such meters are, almost universely, calibrated in terms of rms (to be described in the next paragraph). For pure sine waves, this is quite an acceptable fiction, but for non-sinusoidal waveshapes the reading will be in error.

An alternating current of one ampere rms (or effective) is exactly equivalent to 1 ampere of direct current in heating power as it flows through the resistance of known value. After all, alternating current can heat up a resistor or do work no matter which direction it flows, it is just a matter of evaluating it. In the right hand positive loop of Fig. 1-7 we can read off the ordinates (height of lines to the curve) for each marked increment of time. If we (1) square each value, (2) then add all the squared values together, and (3) find the average, and then (4) take the square root of the average (or mean), we have performed a root-mean-square operation. The point to all this is that power (watts) whether electrical or acoustical is proportional to the square of current or voltage or sound pressure and the rms value is the effective value for the alternating signal. Thank goodness the meters will do this for us, but now we know the significance of the reading.

In Fig. 1-7 is a summary of useful relationships which may

come in handy from time to time. These numerical relationships apply only to the sine waveshape. Ordinarily we just read what the meter says, but it is always desirable to know what is behind that reading.

WAVELENGTH AND FREQUENCY

In Fig. 1-8 a simple sine wave has been plotted against time, representing, for example, the pressure of a sound wave from a tuning fork. We disregard the static atmospheric pressure as our microphone is sensitive only to the rapidly varying part. The *wavelength* is the distance the wave travels in the time it takes to complete one cycle. *Frequency* is the number of cycles per second (hertz). Frequency and wavelength are related as follows:

$$\text{Wavelength (feet)} = \frac{\text{Speed of sound (feet per second)}}{\text{Frequency (hertz)}} \qquad \textbf{(1-1)}$$

which can be written as:

$$\text{Frequency} = \frac{\text{Speed of sound}}{\text{Wavelength}} \qquad \textbf{(1-2)}$$

The speed of sound in air is about 1,130 feet per second (770 miles per hour) at normal temperature. For sound traveling in air, Equation 1-1 becomes:

$$\text{Wavelength} = \frac{1{,}130}{\text{Frequency}} \qquad \textbf{(1-3)}$$

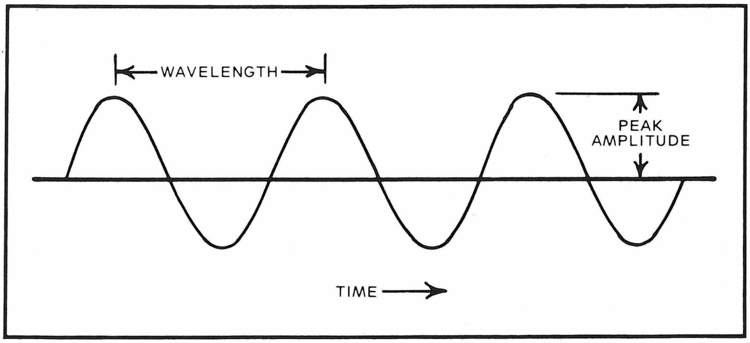


Fig. 1-8. Wavelength is the distance a wave travels in the time it takes to complete one cycle. It may also be expressed as the distance from one point on a periodic wave to the corresponding point on the next cycle of the wave.

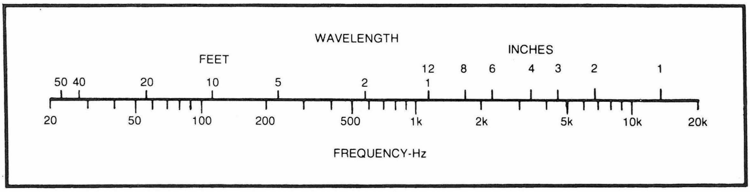


Fig. 1-9. Convenient scales for rough determination of wavelength of sound in air from known frequency, or vice versa. (Based on speed of sound of 1130 feet per second.)

We shall have many occasions to use this relationship. Figure 1-9 is a comparison of a wavelength scale to a frequency scale for ready solution of Equation 1-3. The same information is conveyed in the form of a graph in Fig. 1-10. Some will find one form more convenient to use than the other. These will be useful later when we consider listening room and studio dimensions in terms of the wavelength of sound of various frequencies.

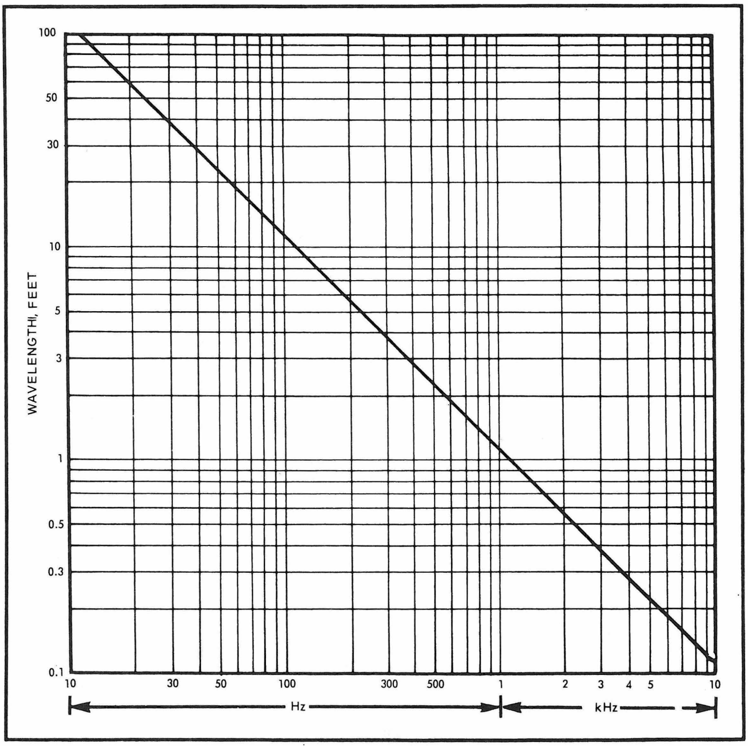


Fig. 1-10. A chart for easy determination of the wavelength in air of sound waves of different frequencies. (Based on speed of sound of 1130 feet per second.)

INVERSE SQUARE LAW

We have all observed the decrease in the intensity of sound as we move away from the source of sound. For those working with recorders and microphones it is of great practical value to be able to predict the change of sound level to be expected with a given change of distance between sound source and microphone or between loudspeaker and listener. To be able to make such predictions requires some knowledge of how sound spreads out.

There are two main reasons for sound intensity decreasing with distance: (1) losses in the medium and (2) geometrical divergence. For modest distances and lower frequencies losses in the air are so small that we shall neglect them and concentrate on geometrical divergence.

If you spread a given amount of butter over a small piece of bread it will be thicker than if spread over a large slice. So it is with sound. All the sound energy radiated by the source S in Fig. 1-11

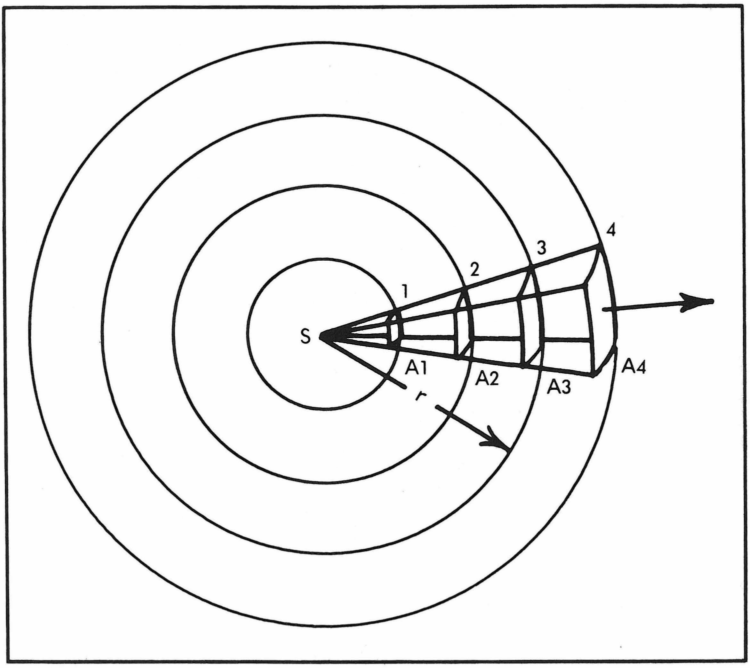


Fig. 1-11. Sound energy radiating uniformly in all directions from a point source must pass through spheres of increasing area. Therefore, the sound power per unit area will vary inversely as the square of the radius of the sphere. Sound pressure falling off according to this "inverse square law" is reduced 6 dB for each doubling of the distance.

must pass through spheres 1, 2, 3, 4, etc. Taking a small solid angle, the same sound power passing through area A_1 also passes through A_4. Because A_4 is of much greater area than A_1, the sound power per square inch at A_4 is much less than at A_1. As the area of a sphere is $4\pi r^2$, the sound power per unit area decreases as the square of the distance. This is called the "inverse square law" and accounts only for the geometrical spreading of the sound and is not a loss in the strict sense of the word.

Anticipating a later discussion on the decibel, this is the proper place to mention an expression of the inverse square law in a very useful form. It can be expressed as "6dB per distance doubled". For example, if a microphone is 5 feet from an enthusiastic soprano and the VU meter in the control room peaks +6, moving the microphone to 10 feet would bring the reading down *approximately* 6dB. The word "approximately" is important. The inverse square law holds only for free field conditions. The effect of sound energy reflected from walls would be to make the change for a doubling of the distance something less than 6dB.

An awareness of the inverse square law is of distinct help in estimating acoustical situations. For instance, a doubling of the distance from 10 to 20 feet would, for free space, be accompanied by the same sound pressure level decrease, 6dB, as for a doubling from 100 to 200 feet. This accounts for the great carrying power of sound outdoors.

REFLECTION OF SOUND

Sound is reflected from objects which are large compared to the wavelength of the impinging sound. This book would be a good reflector for 10 kHz sound (wavelength about an inch). At the low end of the audible spectrum, 20 Hz sound (wavelength about 56 feet) would sweep past the book and the person holding it as though they didn't exist and without appreciable "shadows."

Reflected sound at the higher audio frequencies follows the same rules as light: the angle of incidence equals the angle of reflection as in Fig. 1-12(A). Figure 1-12(B) shows how this law of reflection can be put to good use in a parabolic reflector which focuses incoming parallel rays of sound onto a microphone, a good arrangement for the recording of birdsongs or the marching band at a football game. In the early days of broadcasting sporting events in Hong Kong, a resourceful technician saved the day by placing an ordinary Chinese wok, or cooking pan, behind his microphone to serve as an emergency directional pickup.

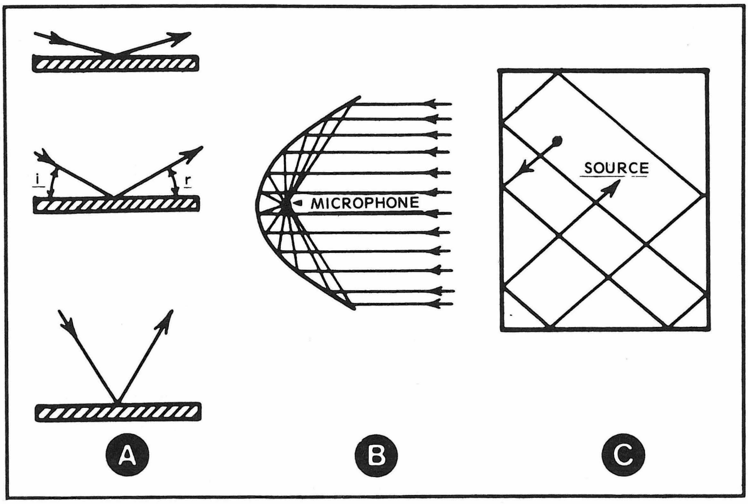


Fig. 1-12. Sound at the higher audio frequencies is reflected from surfaces like light, the angle of incidence (i) equal to the angle of reflection (r). This is true for (A) flat surfaces, (B) parabolic or other curved surfaces, and (C) the walls of listening rooms and studios.

Figure 1-12(C) illustrates how a ray of sound may undergo many reflections in an enclosure such as a studio. How many reflections a single ray would experience depends on how much sound is absorbed at each bounce. It should also be borne in mind that such rays are oversimplifications. They should be considered more as "pencils" of sound with more or less spherical wavefronts which diverge and to which the inverse square law applies.

Considering rays of sound in a studio is helpful up to a point but for a more complete picture of what is happening to our precious music or speech signals we must consider the entire complex sound field. We shall see later how this can be done.

THE STRANGE CASE OF CORNER REFLECTORS

In an art museum with large Dutch master paintings on the wall have you ever been intrigued by the way the eyes of certain subjects seem to follow you accusingly as you walk past? Corner reflectors are like that. There is no escaping their pernicious effect. The corner reflector in Fig. 1-13, receiving sound from a source at S, sends a reflection directly back toward the source. If the angles of incidence and reflection are noted carefully, a source at B emitting a sound still finds a direct, double-surface reflection returning to the source. A source at C, on the opposite of A, experiences the same

effect. We are instinctively aware of perpendicular reflections from surrounding walls, but now, with further enlightenment, we must bear the burden of giving due consideration to reflections from four corners of the room and both of these types of reflections follow the source around the room. Corner reflections suffer losses at two surfaces, hence will tend to be less intense than normal (perpendicular) reflections at the same distance. These are direct reflections. There are many, many indirect reflections involving 2, 3, 4 or more surfaces as we shall see as we study reverberation in Chapter 10.

The corner reflector of Fig. 1-13 is a two-dimensional representation. How about the four upper corners of the room formed by ceiling and walls and another four formed by floor and wall surfaces? Although the geometry gets a bit sticky, the same follow-the-source game is played. In fact, sonar and radar people have long employed targets made of three circular plates of reflecting material assembled so that each is perpendicular to the others.

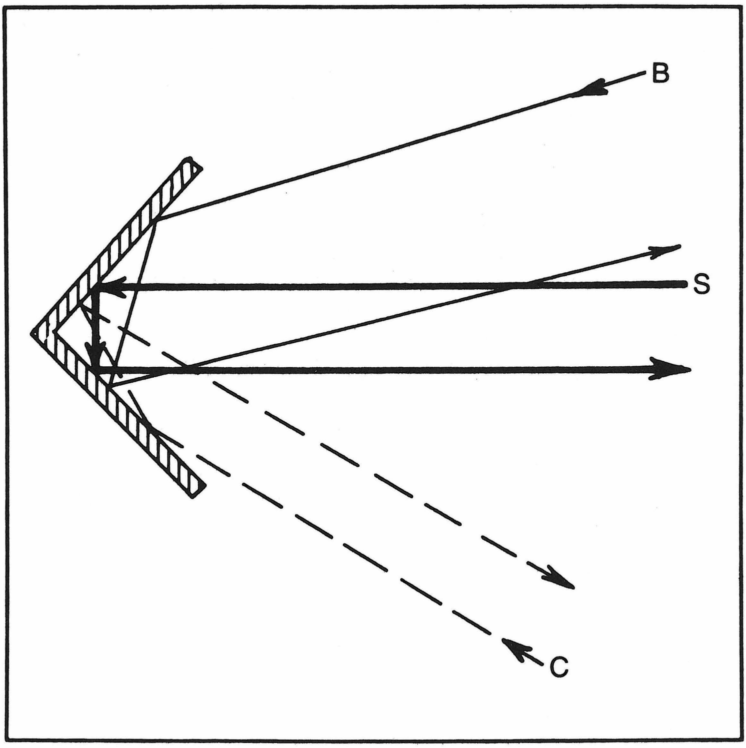


Fig. 1-13. A corner reflector returns sound back toward the source.

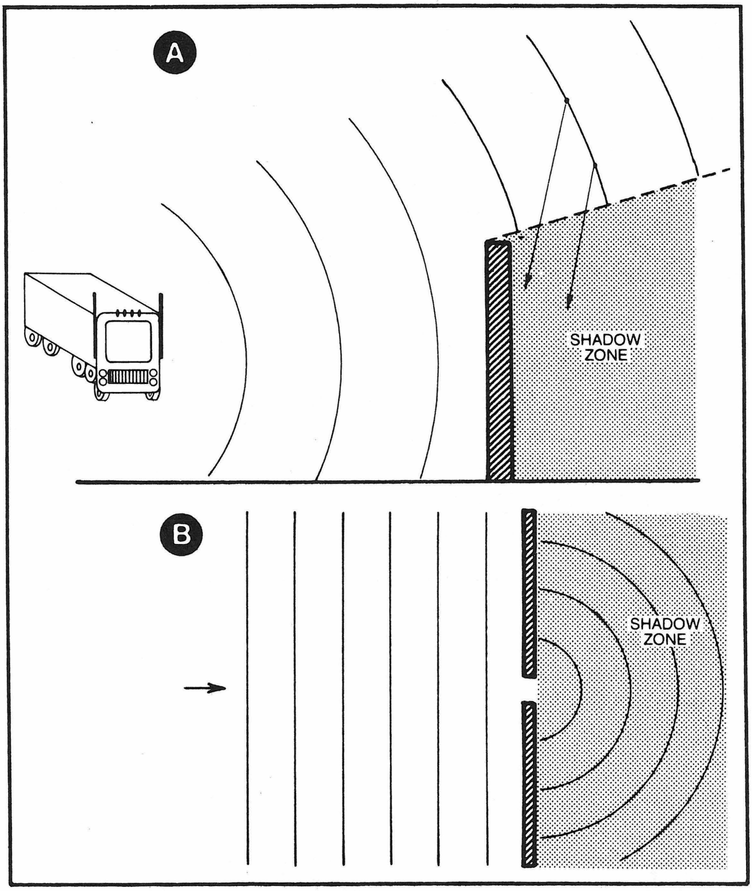


Fig. 1-14. (A) Freeway noise penetrates the shadow zone behind a concrete block wall by diffraction. (B) The shadow zone behind a heavy wall with a hole in it is likewise illuminated by diffracted sound.

DIFFRACTION OF SOUND

Many walls are being built along freeways to protect people from traffic noise. By considering how they work, or rather how they don't work very well sometimes, an understanding of diffraction of sound may be gained. In Fig. 1-14(A) the noise of freeway traffic travels outward as spherical wavefronts. That which hits the heavy concrete block wall is largely reflected back toward the source, making the sound level on the road higher than it would have been without the wall (serves them right!). The region marked "shadow zone" is protected from the direct sound but is penetrated by diffracted sound. Huygen's principle states that every point on

every wavefront at any instant may be regarded as a source of secondary waves traveling out on new spherical wavefronts. These secondary waves, although weaker than the primary waves, send sound energy into this shadow zone.

Figure 1-14(B) illustrates the application of this principle to a plane wave striking a solid wall with a hole in it. In this case the center of the hole can be considered a new source spreading spherical waves into the shadow zone.

We can see that diffraction effects can "bend sound around corners", as it were, pumping sound energy into what we naively considered a black shadow zone. How much sound is bent into a shadow zone depends on the size of the barrier in terms of the wavelength of the sound under consideration.

SUPERPOSITION OF SOUND

Ten people in a room can be looking at ten different objects and the light waves reflected from each object can find their way to the corresponding eye without adverse effect upon the other nine sets of light waves. The principle of superposition of sound waves likewise has important practical results. It means that the air in a room is capable of carrying many sound waves simultaneously. When an orchestra plays, the waves produced by the clarinet do not disturb those produced by the violin. An emergency telephone call from a sound contractor posed an interesting question. In the sound radiating cluster of horns he was installing in a church it was found necessary to cross the axes of two of the horns. He wondered if the sound from one horn would disturb that from another in the air out in front of both. He was assured that by the principle of superposition the little air particles involved would be obedient to the excitation from both horns without problems.

If we toss a stone into a quiet pool of water, a series of concentric waves will emanate from the spot where the stone hits the water. If a second stone is tossed in, a second series of waves will be produced. One series will go through the other with no apparent adverse effect upon either. The principle of *superposition* (or interference) states that the same portion of the medium may transmit simultaneously any number of different series of waves. These proceed independently, each undisturbed by the presence of the others, the displacement of the particles of the medium at any instant being the vector sum of displacements due at that instant and at that point in space to each separate wave system. Let us see what this means in a few specific cases.

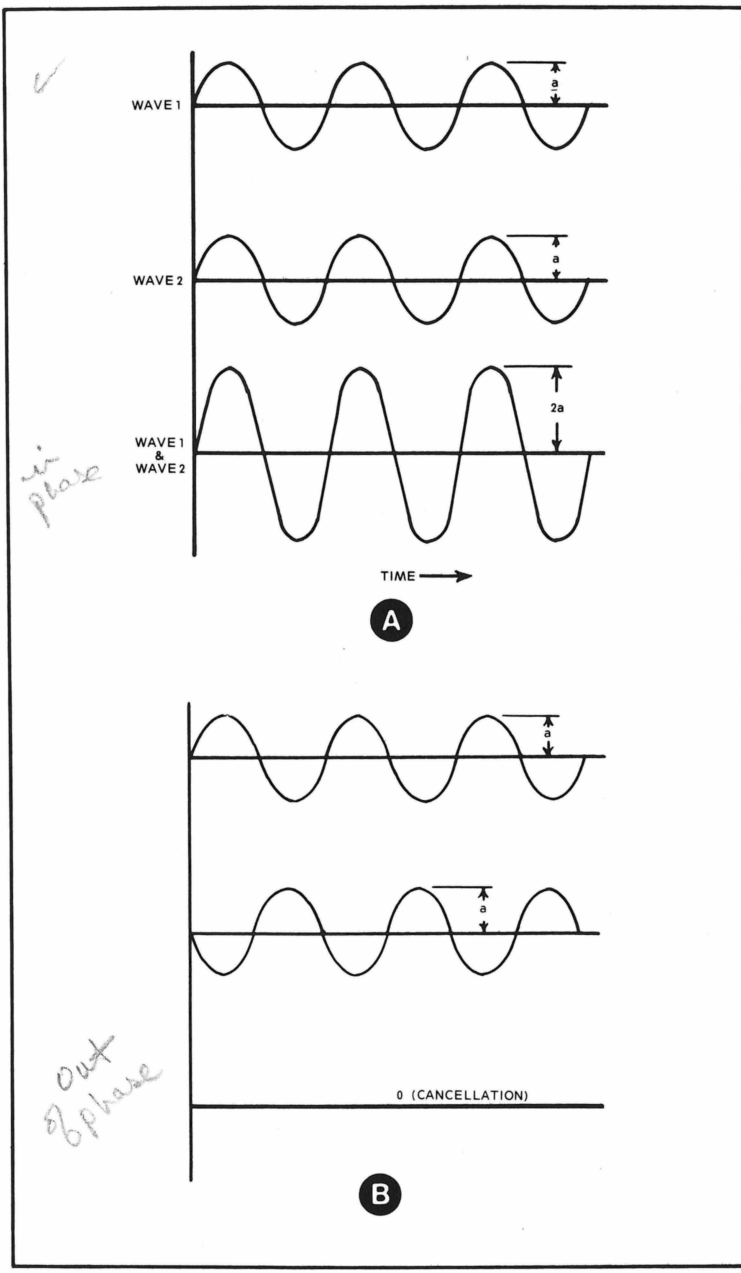


Fig. 1-15. In (A), if wave 1 is superimposed on wave 2 and they are in phase, the amplitude will double. In (B) if wave 1 is a half wavelength (180°) out of phase with wave 2 they will cancel.

In Fig. 1-15(A), waves 1 and 2, arriving from different directions, pass the same spot in the medium at the same time and in phase. The positive and negative peaks of wave 1 add to the peaks of wave 2 giving a resultant having the same frequency but twice the peak amplitude. In Fig. 1-15(B), waves 1 and 2 are of the same frequency and amplitude, but wave 2 arrives later in such a way that the two are in what is called phase opposition. In this case one wave exactly cancels the other and the resultant is zero. For this particular particle, not to move at all is just what is needed to speed both waves 1 and 2 on their respective ways. Of course, there is an infinite number of other combinations of amplitude and phase between two such waves.

If two loudspeakers are placed side by side in a room and energized from the same amplifier driven by a sine wave signal of, say, 500 Hz, these constructive and destructive effects are readily observed by walking around in the resultant sound field. For the moment we are interested primarily in demonstrating the actual physical existence of these super-position effects rather than in the details of the sound field, which would be very complex. In performing this experiment with the two loudspeakers several hints are in order: (a) plug one ear to avoid binaural complications, (b) reflections from walls, ceiling, and floor greatly complicate the field, and (C) a handy device for introducing a phase opposition effect as in Fig. 1-15(B) is to reverse the leads to one of the loudspeakers so that the motion of one cone is out as the other moves in and vice versa. This will shift the location of the maxima and minima in the room.

BENDING SOUND

Sound travels in a straight line as long as it is not reflected and the medium remains uniform. Sound travels at different speeds in different media and at the juncture between two media the direction of travel of the sound is changed. In Fig. 1-16 a wavefront AB in medium 1 is confronted by medium 2 having a lower speed of sound than medium 1. It stands to reason that during the time it takes the wavefront to travel from B to C in medium 1, the associated wavefront at A would travel the shorter distance AD in medium 2 because the sound speed in medium 2 is slower. This means that the wavefront AB and its direction of travel would be changed to that of DC in medium 2 with a new direction of travel. This bending of the direction of travel of a sound ray is called *refraction*.

In the acoustics of rooms, refraction is not very important

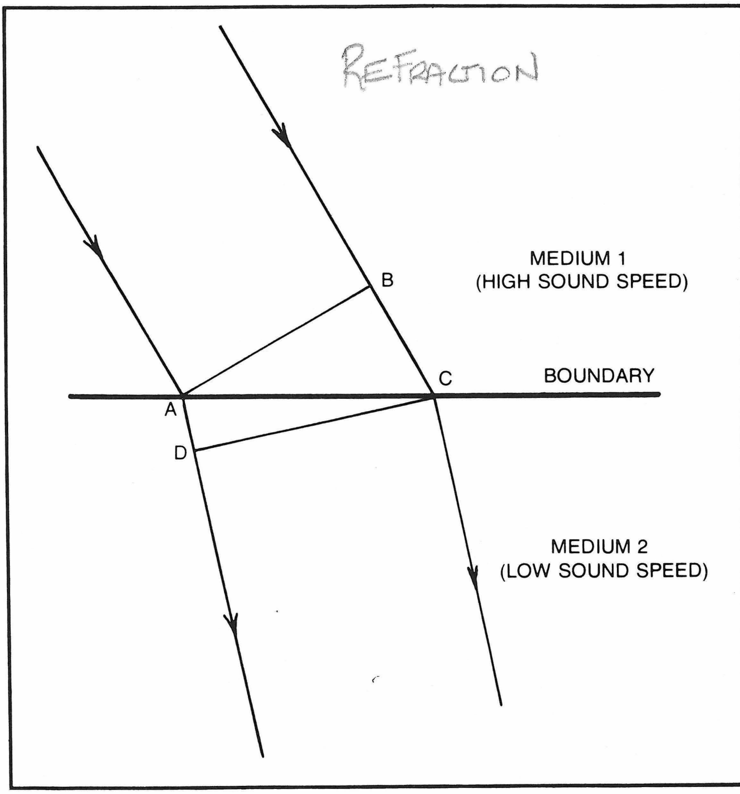


Fig. 1-16. Sound leaving medium 1 and entering medium 2 of lower density will be bent, or refracted.

because we are dealing only with sounds in an air medium. We should, however, be aware that such an effect exists as we list what can happen to sound beams.

ABSORPTION OF SOUND

A reverberation chamber is a room with hard, reflective surfaces. These surfaces are poor absorbers of sound; as a result, the sound decays slowly in such a room. To make a recording studio out of this room would require, among other things, the application of surface materials which would absorb sound efficiently. Absorption of sound is of direct, vital importance to both technical and artistic personnel engaged in high-fidelity recording and reproducing activities. When sound strikes a sheet of absorbing material, as in Fig. 1-17, some of it is reflected back into the room and some of it goes through the sheet, emerging from the other side. We must note two

things, (a) that the intensity of the sound emerging on the other side is lower because of the absorption of sound energy by the sheet and by reflection losses and (b) that in going from the sheet into the air on the other side the sound is refracted back to its original incident direction. As these refraction effects do not affect the intensity of the sound, we may neglect them in this discussion.

The sound energy absorbed by the sheet is changed to heat energy by the friction in the particles in the sheet involved in transmitting the sound. The amount of heat involved is extremely small because the amount of energy involved in normal sounds is very small. In fibrous materials such as carpets, drapes, glass wool, and common acoustical tile, the sound absorption is high (at least at the higher frequencies) because the sound undergoes many reflections in the fibers and tiny pores, losing energy at each reflection. Many commercial acoustical absorbers are made of such fibrous, porous materials.

The fraction of the energy of the incident sound which is absorbed is called the *sound absorption coefficient* of the material. Hard, massive, nonporous materials such as plaster, masonry,

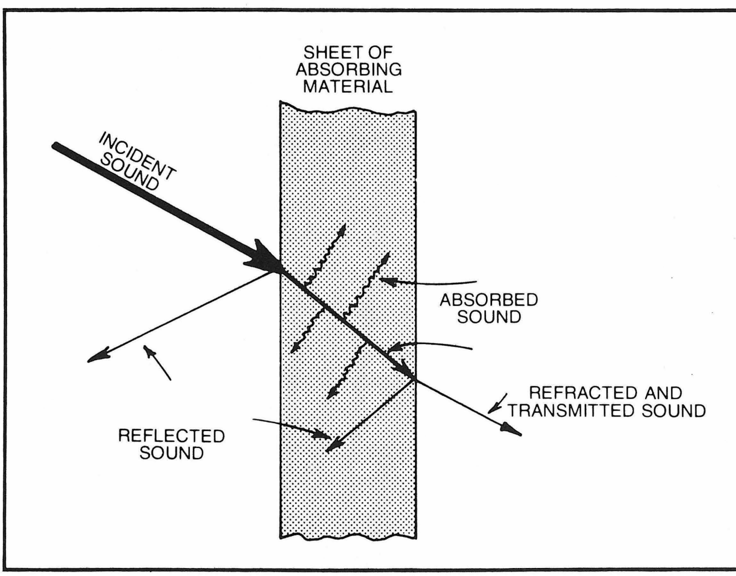


Fig. 1-17. Part of the sound falling on a sheet of absorbing material will be reflected. The part pentrating the sheet will be bent by refraction. Some of the sound energy not absorbed within the sheet will be reflected from the rear boundary of the two media and some will emerge traveling in the same direction as the incident sound.

glass, wood, concrete, etc., have absorption coefficients of less than 0.05, i.e., absorbing less than 5% of the incident energy. Soft, porous materials which permit penetration of the sound waves may have coefficients approaching 1.00 or absorbing close to 100% of the incident energy. We shall consider this more fully in later sections dealing with acoustical materials and studio and listening room design.

COMB FILTER EFFECTS

A special case of superposition of sound is so important as to justify emphasis to help you keep comb filters out of your hair. Phasing and flanging, allied effects well known among contemporary musicians, are done on purpose to create special musical effects. Comb filters in their natural habitat usually come unbidden, creating colorations which distort speech and music, especially speech.

The key word in our investigation of what causes comb filter distortion is *delay*. For example, a direct sound wave of certain amplitude and phase falls on a microphone diaphragm. A reflection of A from a nearby hard surface such as a table top, also falls on the microphone with a slightly lower amplitude and delayed from A because it has traveled farther. These constitute two vectors that combine to provide a resultant which actuates the diaphragm. All the microphone can do is respond to the combination, hence we look more deeply into what happens when a signal is combined with a replica of itself coming along at a slightly later time.

Figure 1-18 shows the effect on the frequency response when a signal, any signal, is combined with a replica of itself delayed 0.1, 0.5, and 1 ms (millisecond). The frequency scale in this case is linear to show the symmetry of the response and to demonstrate how "comb" got into this discussion. We are more familiar with frequency plotted on a logarithmic scale. The conditions of Fig. 1-19 are identical to those of Fig. 1-18, with the response plotted to a logarithmic frequency scale.

In Fig. 1-15(A) we saw that when two in-phase waves of identical frequency and amplitude are combined, the resultant amplitude is doubled. This results in the +6 dB peaks in Figs. 1-18 and 1-19. Figure 1-15(B) shows that when the same waves were in phase opposition, cancellation occurs. These dips go theoretically to minus infinity, practically to −30 or −40 dB. Down through the audible spectrum peaks and valleys appear as shown in Table 1-1.

If the frequency response of our microphone is flat, the re-

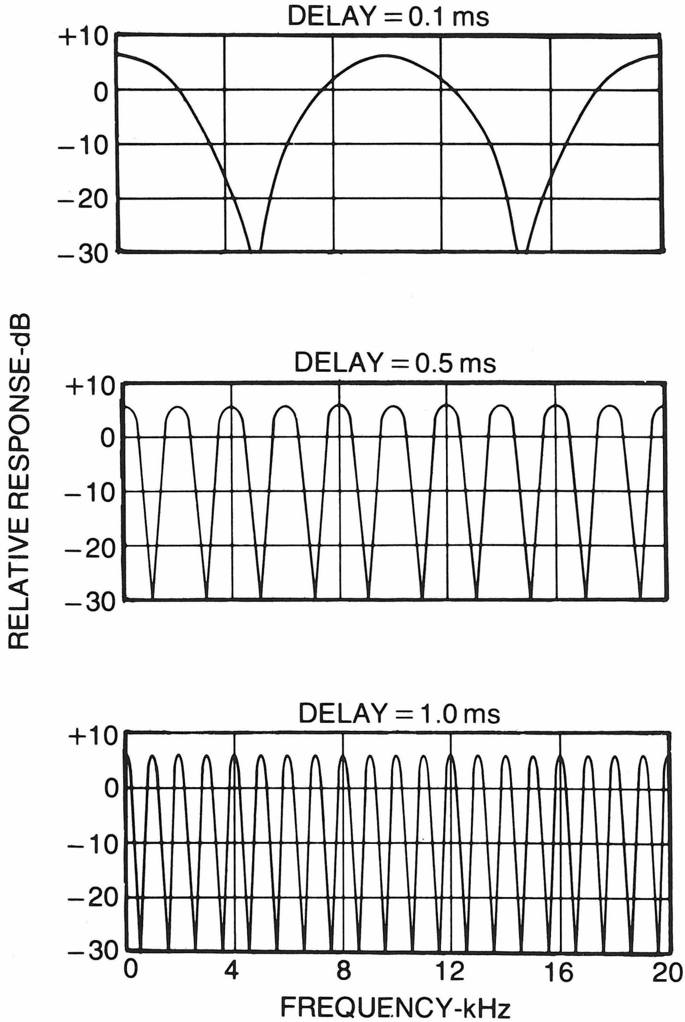


Fig. 1-18. Any speech, music, or other signal combined with itself delayed slightly, will have regions of constructive and destructive interference (superposition) down through the audible spectrum. The effects of three different delays are shown plotted to a linear frequency scale to illustrate the origin of the word "comb" in comb filter. When the undelayed signal is in phase with the delayed signal, double amplitude (+6 dB) results. When in phase opposition, cancellation (−30 or −40 dB) results.

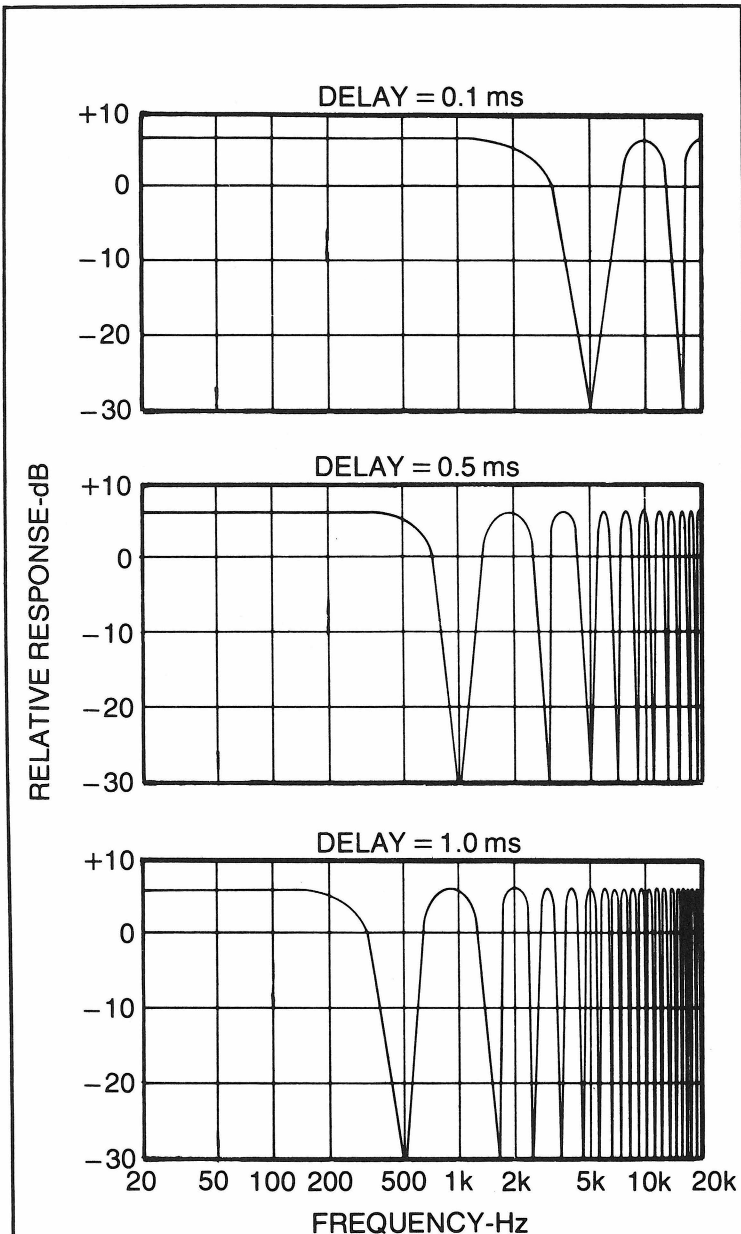


Fig. 1-19. The same condition shown in Fig. 1-18 is here plotted to the more familiar logarithmic scale of frequency.

sponse of our amplifier and all electronics is also flat, our effective, overall response is changed to one of those in Fig. 1-19 by virtue of acoustical combination of a direct signal and an indirect, reflected, delayed component before the signal even gets into the microphone circuit as an electrical signal. The acoustical comb filter effect will appear for further specific considerations in other chapters.

SUMMARY

Are the aspects of sound treated in this chapter of theoretical interest only? Before going on with the study of sound, let us consider a very practical example within the experience of every audio enthusiast and see how many of the principles we have studied come into play.

A speaker emits some beautiful sounds which travel (principle of propagation, inverse square law) to the microphone over the path d_1 as in Fig. 1-20. They also travel via path d_2 which involves a reflection from the tabletop (angle of incidence equals angle of

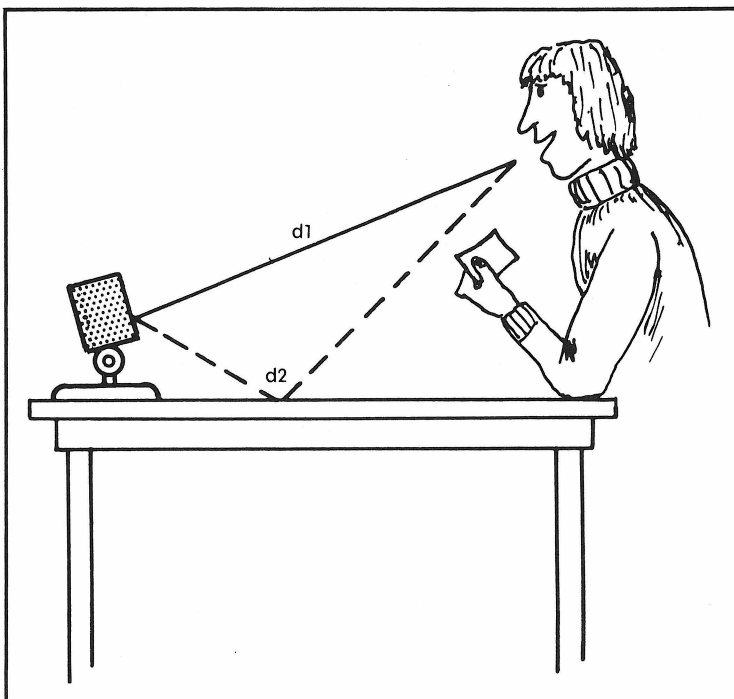


Fig. 1-20. Propagation, inverse square, reflection, and superposition of sound are all involved in the simple act of talking into a microphone.

Table 1-1. Comb Filter Peaks and Dips.

Delay	Frequency of Lowest Dip	Spacing between peaks= Spacing between dips
0.1ms	5,000 Hz	10,000 Hz
0.5	1,000 Hz	2,000
1.	500	1,000
5.	100	200
10.	50	100
20.	10	20

reflection). The sound waves traveling the two routes combine (superposition, comb filter) at the diaphragm of the microphone which, in turn, moves in obedience to the pressure changes in the air in contact with it.

In Fig. 1-15(B) we saw that when waves 1 and 2 were in phase opposition, they canceled each other. Let us take the signal traveling path d_1 of Fig. 1-20 as wave 1 and that bouncing off the table as wave 2. The greater path length of d_2 means that wave 2 arrives later. At some particular frequency, the added distance over path d_2 is just enough to bring wave 2 into phase opposition with wave 1 and the two cancel each other. This means that speech energy near this frequency is lost or seriously reduced. This "destructive" interference occurs at other related higher multiple frequencies according to Table 1-1. In between are frequencies at which the sound pressure is increased ("constructive" interference as in Fig. 1-15(A)). Thus we see that the near presence of the highly reflective surface affects the dulcet tones of the speaker's voice in a most un-hi-fi manner.

This effect may be observed by recording your voice as you hold a microphone a fixed distance from your mouth and then walking toward a hard plaster wall. Playing the test back, a dramatic deterioration of quality is observed as you approached to within a foot or so of the wall. Moral: keep the microphone away from hard, reflective surfaces or reduce the reflectivity of such surfaces (absorbent table cover, rug under the microphone floor stand, etc.). The other approach used in distant pickups is to mount the microphone very close to the floor so that d_1 is essentially equal to d_2 and there is a minimum interference from the reflected component[3].

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2. Sprinkle, Melvin C.. *Acoustics For Audio Men.* db The Sound Engineering Magazine:

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principle of propagation

2

The Language of Decibels

It is no more possible to understand audio apart from a knowledge of the decibel than it is to understand the English language apart from verbs, nouns, and adjectives. Carrying that analogy one step further, a superficial familiarity with the decibel that lacks a basic knowledge of what it is and how it is used in calculations is like being able to identify verbs and nouns without knowing past, present, and future tenses, conjugation of verbs, etc. In other words, a little knowledge of the decibel can be an everpresent barrier that continually trips up the unwary. Therefore, very early in this book, we grapple with the decibel monster to discover that, after all is said and done, the monster turns out to be a marshmallow.

Some people contract a mathematical allergy in early grades from teachers having a profound dislike for the subject. Love and respect for math never came across, but the negative attitude toward it did. Because this allergy has almost reached epidemic proportions among hi-fi aficianados and audio buffs as well as real and aspiring audio engineers, it is time well spent to reveiw decibels. This will bring us all to a common base of understanding which will be new discovery for some, review for others, but helpful to all.

WHY DECIBELS?

The sensitivity range of the human hearing mechanism (to be taken up in the next chapter) is so great that we need some way to

why DECIBELS?

"crunch" the numbers. Decibels do that. We can hear everything from a pin dropping to a rocket blastoff. When the sound pressures associated with these two events are compared, a range of many millions to one is involved. In quite ordinary problems in acoustics very small and very large numbers continually arise and handling them with decibels can be done with grace and facility but without them, simple steps become clumsy, awkward, and inconvenient.

There is another reason for using decibels that has nothing whatever to do with our convenience. Decibels are closely related to the way our senses act. Vision, hearing, sensing of vibration, or even feeling electric shock, all are better described in terms of ratios than differences. For example, a weak sound with an intensity of 1 unit sounds twice as loud with an intensity of 10 units. A loud sound having an intensity of 10,000 units must be increased to 100,000 units to sound twice as loud. In both cases the doubling of loudness is assocated with ratios of 10 : 1. Differences in intensity associated with the doubling of loudness was $10-1=9$ in one case and $100,000-10,000=90,000$ in the other. Ratios seem to fit our sensory responses far better than differences. Decibels and ratios go hand in hand, hence decibels have found wide and effective application in any sort of measurements associated with human hearing.

Table 2-1. Equivalent Number Systems.

Decimal System X	Arithmetic System	Exponential System	Decibel System*
1,000,000	10×10×10×10×10×10	10^6	120 dB
100,000	10×10×10×10×10	10^5	100
10,000	10×10×10×10	10^4	80
1,000	10×10×10	10^3	60
100	10×10	10^2	40
10	10×1	10^1	20
1	10/10	10^0	0
0.1	1/10	10^{-1}	−20
0.01	1/(10×10)	10^{-2}	−40
0.001	1/(10×10×10)	10^{-3}	−60
0.0001	1/(10×10×10×10)	10^{-4}	−80
100,000	(10)(10,000)	$(10^1)(10^4) = 10^5$	100
1,000	100,000/100	$10^5/10^2 = 10^3$	60
10	1000/100	$10^3/10^2 = 10^1$	20
3.16	$\sqrt{10} = {}^2\sqrt{10}$	$10^{0.5} = 10^{1/2}$	10
2.15	${}^3\sqrt{10}$	$10^{1/3}$	6.67
3.74	${}^5\sqrt{27^2}$	$27^{2/5}$	11.45

* Simply 20 log x

WAYS TO HANDLE NUMBERS

Table 2-1 lists different ways that numbers can be represented. The decimal, arithmetic, and exponential systems are equivalent forms of representing a given number. While equivalent in value, the different systems are not necessarily equivalent in convenience. For example, the number 100,000 in the decimal form can be compressed to 10^5 (ten to the fifth power) in the shorthand exponential form. Multiplying (1,000,000) (10,000) (0.001) is rather messy in the arithmetic form with all those zeros to keep track of, but it becomes quite manageable in the exponential form: $(10^6)\,(10^4)\,(10^{-3})=10^7$. That is, it is quite manageable providing you know the simple rules of exponents which are these: (a) in multiplication, exponents add algebraically as in the previous example, (b) in division, exponents are subtracted (example: $10^6/10^4=10^{6-4}=10^2$, another example: $10^6/10^{-4}=10^{(6)-(-4)}=10^{6+4}=10^{10}$). Prefixes used to designate powers of ten in the exponential systems are listed in Table 2-2.

LOGARITHMS

Logarithms are a natural outgrowth of the exponential column of Table 2-1. The number 1,000,000 can be expressed in exponential form as 10^6. The logarithm of 10^6 to the base 10 is 6 and the logarithm of other numbers are:

$$\begin{array}{rcllll}
1 & = & 10^0, & \text{therefore} & \log 1 & = 0 \\
10 & = & 10^1, & \text{therefore} & \log 10 & = 1 \\
100 & = & 10^2, & \text{therefore} & \log 100 & = 2 \\
1{,}000 & = & 10^3, & \text{therefore} & \log 1{,}000 & = 3 \\
1{,}000{,}000 & = & 10^6, & \text{therefore} & \log 1{,}000{,}000 & = 6
\end{array}$$

Everything goes smoothly until we encounter a need to find the log of an odd number such as 876. Gone are the days of having to consider characteristics and mantissas and entering log tables to find the log of 876. The calculator does this simply by entering 876, pressing the log key, and reading off 2.94. The number 876 falls between 100 and 1,000 in the above tabulation and the log of 876 = 2.94 also falls between 2 and 3, thus everything checks out and we conclude that $876=10^{2.94}$ and the log of 876 is 2.94.

The logarithms described above are common logarithms to the base 10. Theoretically, any base can be used, but in audio base 10

Table 2-2. Exponential Prefixes.

Multiple	Prefix	Symbol
10^{12}	tera	T
10^{9}	giga	G
10^{6}	mega	M
10^{3}	kilo	k
10^{-3}	milli	m
10^{-6}	micro	u
10^{-9}	nano	n
10^{-12}	pico	p
10^{-15}	femto	f
10^{-18}	atto	a

dominates. The abbreviation "log" applies to logarithms to the base 10. The abbreviation "ln" applies to logarithms to the base e=2.7183 which is extensively used in the more mathematical treatments of wave motion and vibration. Both are found on the usual engineering calculator.

The beauty of logarithms is that they reduce arithmetic operations to simpler forms. For example , (a) division is reduced to subtraction:

$$\log \frac{A}{B} = \log A - \log B$$

(b) multiplication is reduced to addition:

$$\log (A)(B) = \log A + \log B,$$

and (c) raising a number to a power is reduced to multiplication:

$$\log A^{B} = B \log A.$$

The obsolete slide rule multiplied and divided by adding and subtracting on logarithmic scales.

THE DECIBEL

With this background in number systems and logarithms we are prepared to enter into the mysteries of the decibel column of Table 2-1. The bel is named after Alexander Graham Bell, the

inventor of the telephone. The bel is defined as the logarithm to the base 10 of the ratio of two powers, W_1 and W_2:

$$L = \log \frac{W_1}{W_2} \text{ in bels} \tag{2-1}$$

It is important to emphasize that a ratio of two **powers** is involved in the basic definition, not ratios of pressures, voltages, or what-have-you. The bel, being inconveniently large, is replaced in common practice by the decibel, equal to one tenth of a bel. The level in decibels then becomes:

$$L = 10 \log \frac{W_1}{W_2} \text{ in decibels} \tag{2-2}$$

In acoustics, sound power is proportional to the square of sound pressure for a plane, progressive wave. Since sound pressure is the parameter easily measured:

$$W_1 \sim p_1^2$$

$$W_2 \sim p_2^2$$

and Eq. (2-2) becomes:

$$L = 10 \log \frac{p_1^2}{p_2^2}$$

$$L = 20 \log \frac{p_1}{p_2} \tag{2-3}$$

This is our first encounter with the 10 log and 20 log form which is confusing to many audio workers. The sound power form of Eq. 2-2 is equivalent to the sound pressure form of Eq. 2-3 for the usual plane, progressive wave. In electrical terms the W_1 and W_2 of Eq. 2-2 are in watts and may be expressed as:

$W = E^2/R$

$$W_1 = \frac{E_1^2}{R} = I_1^2 R$$

$W = I^2 R$

$$W_2 = \frac{E_2^2}{R} = I_2^2 R$$

$W = EI$

$E = IR$

where

E_1 and E_2 are voltages.

I_1 and I_2 are electrical currents, and
R = resistance of the circuit.

In terms of electrical voltage:

$$L = 10 \log \frac{W_1}{W_2}$$

$$L = 10 \log \frac{\frac{E_1^2}{R}}{\frac{E_2^2}{R}} = 10 \log \frac{E_1^2}{E_2^2}$$

$$L = 20 \log \frac{E_1}{E_2} \qquad (2\text{-}4)$$

In terms of electrical current:

$$L = 10 \log \frac{I_1^2\ R}{I_2^2\ R} = 10 \log \frac{I_1^2}{I_2^2}$$

$$L = 20 \log \frac{I_1}{I_2} \qquad (2\text{-}5)$$

Whenever power (acoustical, electrical, or other) is involved, the 10 log form applies. When sound pressure or electrical current or voltage are involved, the 20 log form applies because power is proportional to the square of these parameters. Basically, though, the decibel is a logarithm of a ratio of powers.

REFERENCE LEVELS

Equation 2-3 states that the level is 20 log of a ratio of two pressures p_1 and p_2. In Table 2-1, the right hand decibel column is

Table 2-3. Some Common Reference Levels.

Quantity	Symbol	Formula	Reference Level
Sound Pressure Level	L_p	$20 \log(p_1/p_2)$	$p_2 = 20\,\mu Pa$ (in air)
Audio Power Level	dBm or VU	$10 \log(W_1/W_2)$	$W_2 = 10^{-3}$ watt (1mw)
Sound Power Level	L_w	$10 \log(W_1/W_2)$	$W_2 = 10^{-12}$ watt (1 picowatt)
Voltage	dBv	$20 \log(V_1/V_2)$	$V_2 = 1$ volt

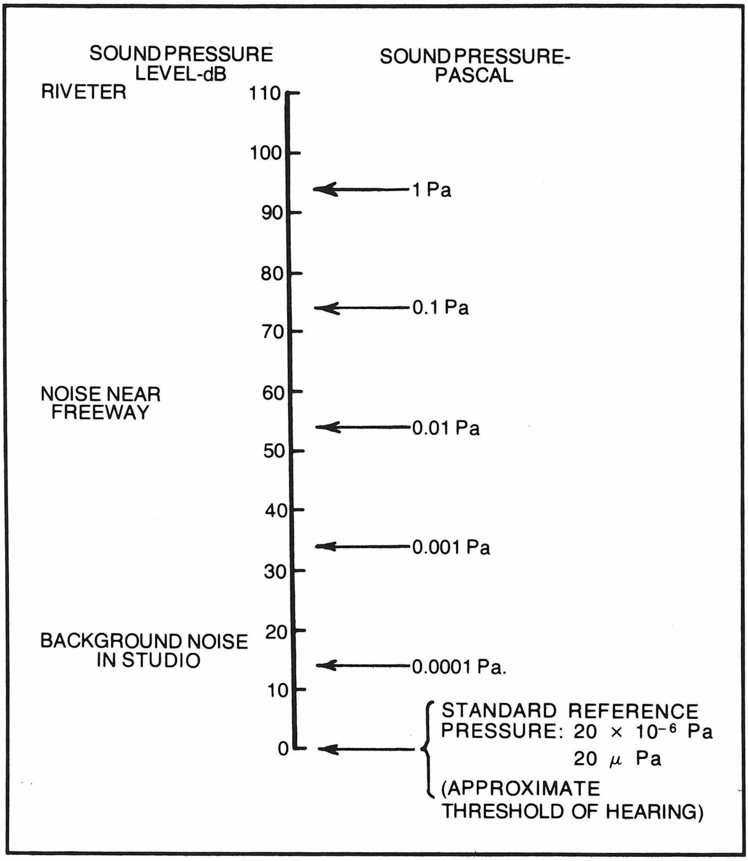

Fig. 2-1. An appreciation of the relative magnitude of a sound pressure of 1 Pascal may be gained by comparison to known sounds. The standard reference pressure for sound in air is 20 μ Pa which corresponds closely to the minimum audible pressure.

simply 20 log of the left hand column. In other words, p_1 is the value in the left hand column and p_2, the reference quantity in this case is equal to unity. This serves to compare the wild range of the decimal form to the more tractible range of the decibel form. However, the reference quantity, p_2, does not equal 1 in normal acoustical problems. Rather, the standard reference sound pressure agreed upon internationally is 20 μPascal, a very small sound pressure close to the audible threshold of the human ear. To obtain some physical appreciation of a pressure of 20 μPa magnitude, Fig. 2-1 is presented. Wherever this standard reference pressure is used, the L in Eq. 2-3 then automatically becomes a *sound pressure level* and

whenever these three words occur together, one can assume that the p_2 reference pressure is 20 μPa. The definition of the Pascal is: a unit of pressure corresponding to a force of 1 newton acting uniformly upon an area of 1 square meter, hence 1 Pa = 1 N/m^2.

LOGARITHMIC AND EXPONENTIAL FORMS COMPARED

The logarithmic and exponential forms are equivalent as can be seen by glancing again at Table 2-1. In working with decibels it is imperative that a familiarity with this equivalence is firmly grasped.

Let us say we have a power ratio of 5:

$$10 \log_{10} 5 = 6.99 \text{ is exactly equivalent to}$$

$$5 = 10^{\frac{6.99}{10}} \tag{2-6}$$

There are two tens in the exponential statement, but they come from different sources as indicated by the arrows. Now let us treat a sound pressure ratio of 5:

$$20 \log_{10} 5 = 13.98$$

$$5 = 10^{\frac{13.98}{20}} \tag{2-7}$$

The utility of this equivalence can be demonstrated with a few examples.

Example 1: A sound pressure level of 73 dB is measured with a sound level meter. What sound pressure does this level represent? From Eq. 2-3:

$$20 \log_{10} \frac{p_1}{p_2} = 73 \text{ dB}$$

$$\frac{p_1}{20 \times 10^{-6}\text{Pa}} = 10^{\frac{73}{20}} = 4{,}466.8$$

$$p = (4{,}466.8)(20 \times 10^{-6}) = 0.0893 \text{ Pa}$$

The $10^{\frac{73}{20}} = 10^{3.65}$ can be evaluated on the calculator by the y^x key, taking y=10 and x=3.65. For a rough and ready solution to this the graph of Fig. 2-2 can be used. As a check, entering the graph with the sound pressure of 0.0893 Pa, 73 dB is read off the left scale confirming the calculations.

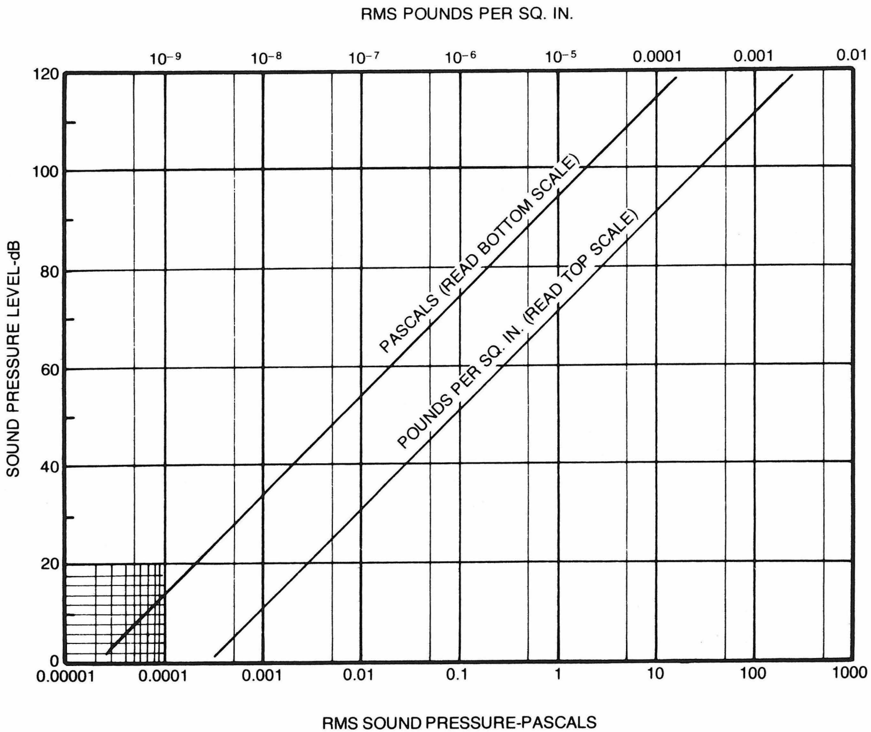

Fig. 2-2. The relationship between sound pressure in Pascals or pounds per square inch and sound pressure level (referred to 20 μPa) is shown in this graph. This is a graphical approach to the solution of Eq. 2-7.

Example 2: The sensitivity of a loudspeaker is given as 82.5 dB sound pressure level at a distance of 10 ft. with 1 watt input. What would the sound pressure level be at the same distance if the power to the loudspeaker were increased to 15 watts?

$$82.5 + 10 \log \frac{15}{1} = 82.5 + 11.76 = 94.26 \text{ dB}$$

This provides a good observation lesson that decibels are decibels. The 82.5 dB figure is a sound pressure level. Increasing the power to the loudspeaker obviously increases the sound pressure level. How much increase is determined by 10 log of a power ratio and the dB increment increase can be added to the 82.5 dB.

COMBINING DECIBELS

As defined in Eq. 2-2, the decibel is always a logarithm of a ratio of two powers. We may use other parameters such as pressure or voltage to figure our decibels, but remember that the 20 log factor keeps the basic power concept intact as pressure squared and voltage squared are proportional to power. The combining of decibels must always be done on a power basis.

Example 1. Let us say it is warm in our studio and a fan is brought in to augment the air conditioning (A/C) system. If both fan and A/C are turned off a very low noise level prevails, low enough to be neglected in the calculation. If the A/C alone is running, the sound pressure level at a given position is 55 dB. If the fan alone is running the sound pressure level is 60 dB. What will be the sound pressure level if both are running at the same time?

$$\begin{aligned} \text{Combined dB} &= 10 \log \left(10^{\frac{55}{10}} + 10^{\frac{60}{10}}\right) \\ &= 61.19 \text{ dB} \end{aligned}$$

Example 2. If the combined level of two noise sources is 80 dB and the level with one of the sources turned off is 75 dB, what is the level of the remaining source?

$$\begin{aligned} \text{Difference dB} &= 10 \log \left(10^{\frac{80}{10}} - 10^{\frac{75}{10}}\right) \\ &= 78.3 \text{ dB} \end{aligned}$$

In other words, combining the 78.3 dB level with the 75 dB level gives the combined level of 80 dB.

Example 3. In a control room a random noise applied to the left loudspeaker gave a sound pressure level at the operator's position of 88 dB. The same noise signal applied to the right loudspeaker alone also gave a sound pressure level at the same position of 88 dB. If both loudspeakers are energized at the same time, what is the combined sound pressure level at the operator's position?

$$\begin{aligned} \text{Combined dB} &= 10 \log (10^{\frac{88}{10}} + 10^{\frac{88}{10}}) \\ &= 91.01 \text{ dB} \end{aligned}$$

Doubling the power results in a 3.01 dB increase. A simpler approach to this problem would have been 10 log 2 = 3.01 dB.

If both loudspeakers are driven by the same noise generator, the signals are coherent. For this example, the second loudspeaker should really be driven by a second noise generator so that random relationships prevail.

Example 4. Scottish bagpipers love the old tune, "We're a hundred pipers in a′ and a″". If a single piper produced a sound pressure level of 65 dB at a given distance, what sound pressure level would be produced by 100 of them tooting away with the breeze through their knees?

$$\begin{aligned} \text{Total Sound Pressure Level} &= 65 + 10 \log 100 \\ &= 65 + 20 \\ &= 85 \text{ dB} \end{aligned}$$

This brings up an important distinction between random and coherent sounds. The previous control room example illustrates the random case in which the amplitude and phase of the two signals are completely random and unordered. Figure 1-15 illustrates the combination of two waves of the same frequency, amplitude, and in phase which yields a doubling of the amplitude, hence a 6 dB increase. On a power basis this would be $10 \log (2)^2 = 10 \log 4 = 6.02$ dB. It is inconceivable that 100 pipers would be so completely in unison that their 100 frequencies, amplitudes, and phases would add in a coherent way.

3

Speech, Music, and Noise

The ultimate goal in the design and use of high fidelity recording and reproducing systems is to handle speech and music signals without altering them appreciably. Engineering genius has made available recording and reproducing systems of remarkable performance, but what comes out of the system depends on two things, (1) what goes in and (2) what happens to it as it passes through the system. In this chapter we deal with the complexities of the signals applied to the input of the system and something of the types of distortion which could occur within, but the rest we leave to the electronics engineer. Speech, music (except synthetic), and some noise originates in acoustical space with which the book is primarily concerned.

SPEECH

The energy involved in speech is amazingly low. A normal conversational voice registering 65 dB (A-weighting) at a distance of 3 feet from the person speaking has an average power of 50 millionths of a watt. That is, it would take the simultaneous power of a million such conversationalists to light a 50-watt electric lamp! However, lighting lamps with hot air is not as interesting to the hi-fi specialist as is the distribution of speech power with frequency.

As much as 95 percent of speech power is concentrated below 1000 Hz and yet the other 5 percent is of extreme importance in understanding speech. As we are confronted with tone controls and

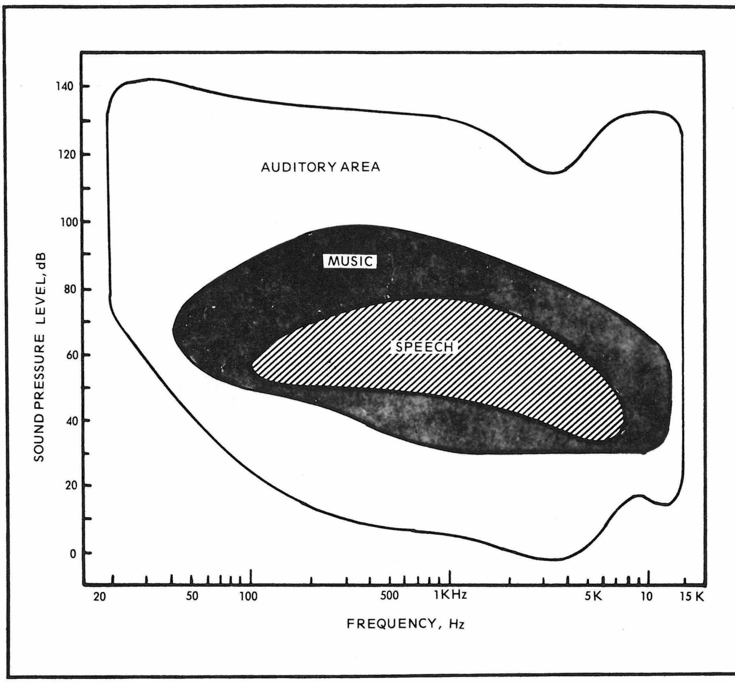

Fig. 3-1. Speech utilizes a relatively small portion of the auditory area, music considerably more. These two areas indicate the frequency range (horizontal scale) and the dynamic range (vertical scale) required for speech and music.

equalizer knobs in a speech circuit, it is well to remember that, in general, most of the power of speech is in the low frequencies. Good intelligibility is dependent upon the presence of that very important portion of power in the highs.

What is required for high fidelity reproduction of speech? Certainly a good signal-to-noise ratio and freedom from distortion are assumed. Beyond this, near-perfect understandability and voice recognition might be listed. All of these required features are the very fabric of high fidelity techniques. However, good high frequency response can create problems. For instance, some voices have an overabundance of sibilants, sometimes caused by dentures, sometimes a natural result of a particular mouth shape, tongue, and set of teeth. These high frequency sibilants may require some "de-essing" or rolling off of the highs to bring them under control. Because of the greater directivity of the sibilants, having a speaker direct his or her voice to one side of the microphone may achieve similar results.

In Fig. 3-1 we see that speech occupies only a small fraction of

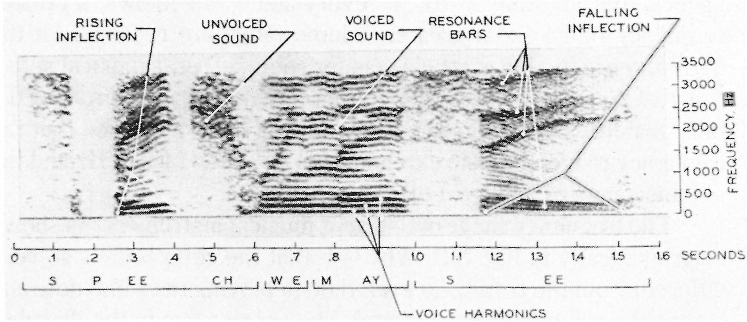

Fig. 3-2. A sound spectrograph analysis reveals how complicated speech sounds really are. Intensity is indicated by the darkness of the tracing. (Courtesy Bell Laboratories)

the total auditory area. Studying this speech area we see that a frequency range from about 100 to 8000 Hz is sufficient for reproduction of speech with perfect fidelity. What happens if our system frequency range is narrower than this? It has been found that good understandability of speech (high articulation) can be obtained with a surprisingly narrow frequency range, although, of course, at the expense of quality. The telephone is a good example of this. Practical communication and fair speaker identification is accomplished in telephone systems with a frequency range of only about 200 to 3000 Hz.

The volume range required for faithful reproduction of speech can also be found by inspecting the speech area of Fig. 3-1. As the speech area extends from about 35 dB on the bottom to about 75 dB on the top, the volume range is thus 40 dB.

Many sounds, including speech, vary with time in a distinctive way. The distribution of energy in such sounds may be studied with a special "three-dimensional" analyzer displaying variations in intensity with both frequency and time. Such a device is called a "sound spectrograph" which is the instrument used to produce "voiceprints" sometimes admitted in courtrooms to establish who spoke certain recorded words. Figure 3-2 shows a spectogram of a human voice saying, "Speech we may see." The frequency increases toward the top of the figure; time progresses from left to right; the higher the intensity the blacker the trace. Notice especially the continuous-type spectra for the "s" and "ch" sounds.

MUSICAL SOUNDS

The musical area of Fig. 3-1 is considerably greater than the

speech area. In other words, as every audiophile knows, a greater frequency range and a greater volume range are required for the faithful reproduction of music than for speech. Yet, a musical selection that demands all of the music area comes far from exercising the full capability of the human ear. From Fig. 3-1 we can see that the frequency range of music extends from 40 Hz to 14,000 Hz and the volume range is about 70 dB.

The frequency range of different musical instruments is shown diagramatically in Fig. 3-3. Why is it that the cello sounds so very different from the trombone even though playing the same note and having the same frequency range? The answer lies in the distribution of energy throughout this range. All of these instruments produce line spectra of the type to be considered later and the relative intensity and phase of the harmonics and overtones are responsible for the distinctiveness (timbre) of each.

OTHER COMMON SOUNDS

It is easy to get so wrapped up in speech and music that other common sounds in life are overlooked or neglected. If we are really interested in audio and acoustics we should also be consciously

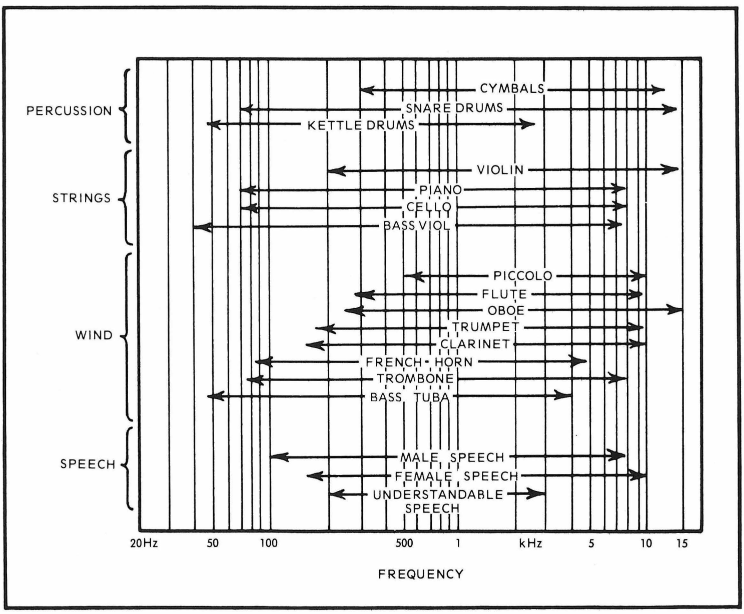

Fig. 3-3. The frequency range of speech and common musical instruments. Two instruments having the same frequency range may sound quite different because of difference in harmonic structure.

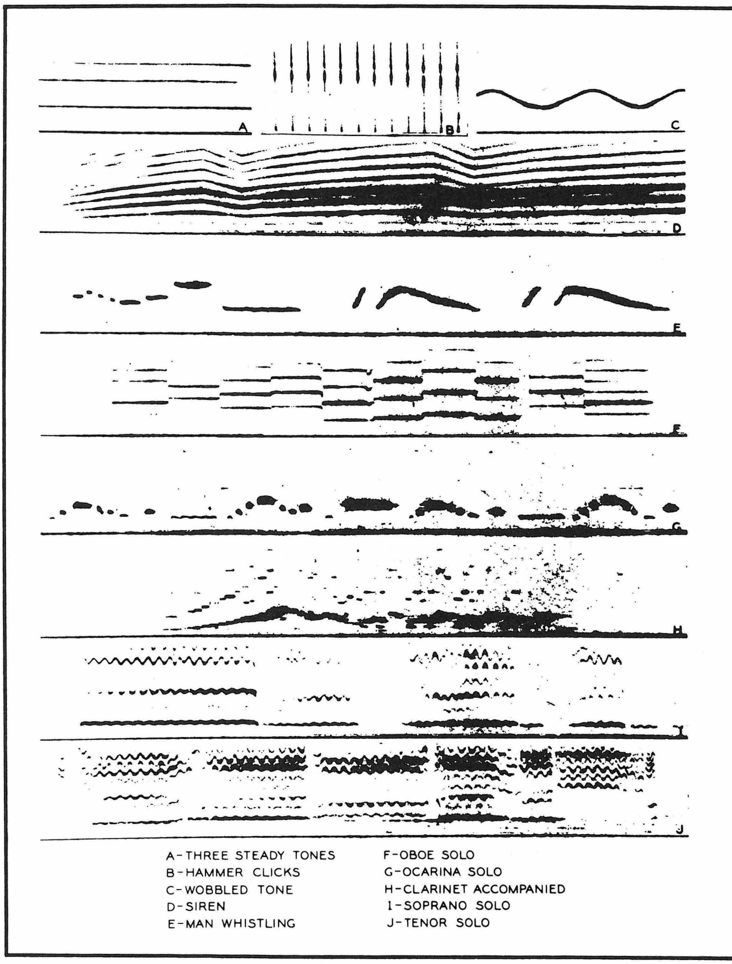

Fig. 3-4. Sound spectograms of the following: tones, clicks, siren, whistling, oboe, ocarina, clarinet, soprano voice, and tenor voice. (Courtesy Bell Laboratories)

aware of the very common sounds that envelop us throughout the day, which give us a sense of being alive and a part of our environment. Even as speech and music, all of these common sounds must also be played out on the auditory area of Fig. 3-1.

As we listen objectively to a common sound it is instructive to consider it from the technical viewpoint. What frequency range does it cover? Is it tonal in nature or rich in harmonics? What changes occur with time? Listening critically and analytically to a bird's song need not degrade our appreciation of its beauty, it can

enhance it. A physicist viewing a sunset can appreciate its beauty more fully knowing that the red color predominates because light of longer wavelength (the red) is absorbed less in the atmosphere than the shorter wavelengths. He sees not only the esthetic beauty, but also the beauty of the mechanism involved.

The sound spectrographic analysis of Fig. 3-2 of the words,

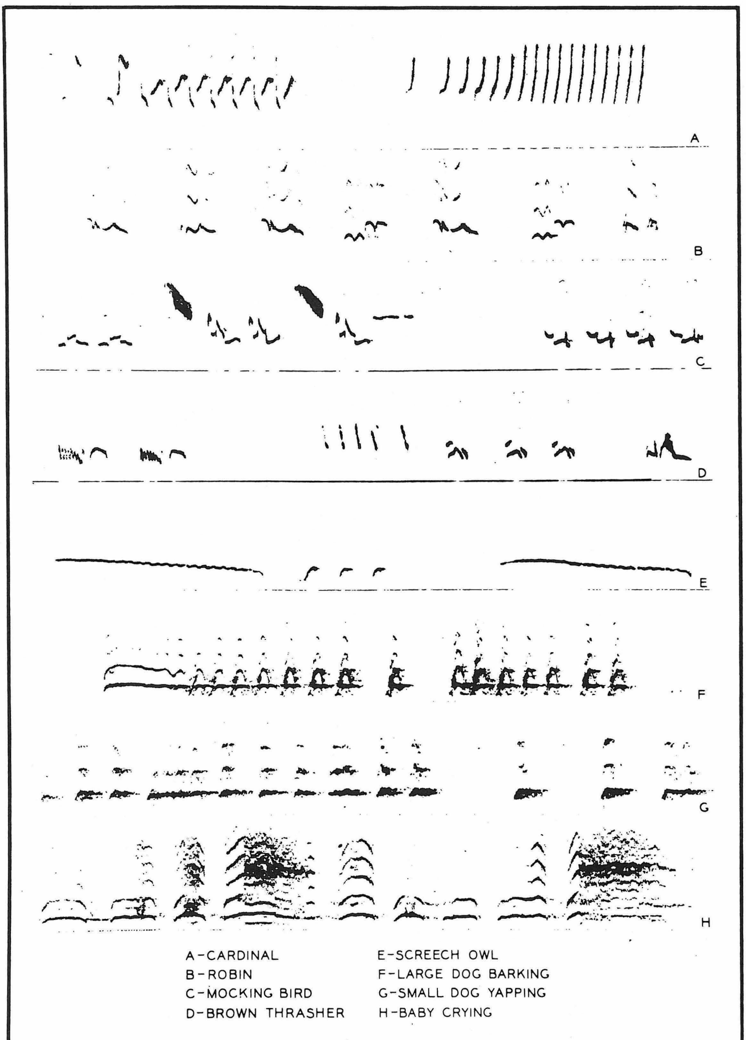

Fig. 3-5. Spectograms of the following sounds: cardinal, robin, mocking bird, brown thrasher, screech owl, large dog barking, small dog yapping, and baby crying. (Courtesy Bell Laboratories)

"Speech we may see," gives a vivid appreciation of the complexity of speech. Such an analysis of common environmental sounds allow us to evaluate at a glance their spectral content and changes with time as well. Figure 3-4(A) shows steady tones of three different frequencies; steady because they are uniform with time; 3 frequencies because there are three horizontal lines displaced on the vertical frequency scale. The hammer clicks of Fig. 3-4(B) are equally spaced in time (horizontal scale) and the density of each vertical line gives some idea of how the energy is distributed in frequency (vertical scale). A varying tone is shown in Fig. 3-4(C) going up and down in frequency as time progresses. So much for illustrating the principles involved. The siren of Fig. 3-4(D) can almost be reproduced mentally by seeing the parallel bands which say, "a fundamental and many harmonics", and the upward sloping lines saying, "pitch increasing", and the downward sloping lines saying, "pitch decreasing". The time scale is not shown, hence we do not know whether it is one of the wailing sirens or one of the "wah-wah-wah" type! With such an analytical attitude, try to conjure up a mental picture of the other sounds of Figs. 3-4 and 3-5.

DYNAMIC RANGE

The dynamic range of a device is the usable intensity range bounded at the lower extreme by noise and at the upper extreme by distortion. In Fig. 3-1 we see that the ear can effectively handle a tremendous range of sound intensities between the threshold of hearing and the threshold of feeling. At a frequency of 1000 Hz this range approaches 130 dB. The equipment man has built does not match the ear in dynamic range.

Every device (such as tape and cassette recorders, phonograph reproducers, amplifiers, transmitters, radio receivers, etc.) generates internal noise which sets a lower limit on signals that can be handled without being dominated by the noise. The noise may be tape hiss, record surface noise, atmospherics, man-made interference, the noise produced by the thermal agitation of electrons flowing in conductors, power circuit hum, or a combination of these. It is interesting that the thermal noise level of air molecules acting on the diaphragm of a very sensitive microphone is of the same order of magnitude as the thermal noise level in electrical conductors. Thus the smallest signal that can be handled by a communication system is fixed by the ever-present noise.

Every recording and reproducing device is also limited in how strong a signal it can handle. This upper limit of signal strength is

Table 3-1. Peak Power of Musical Instruments.

Instrument(s)	Whole spectrum peak power, watts
Bass Drum (36″×15″)	24.6
Snare Drum	11.9
15-Piece Orchestra	5.3 - 12.7
75-Piece orchestra (microphone near conductor)	8.2 - 66.5
Pipe Organ	2.5 - 12.6
Trombone	6.4
Trumpet	0.314
French Horn	0.053
Clarinet	0.050
Flute	0.055
Piccolo	0.084
Piano, 12 ft	0.437
Bass Viol	0.078
Tuba – BB^b	0.082
Bass Saxophone	0.228

(From Sivian, Dunn, and White[1])

set by distortion. As the signal level is increased, a point is eventually reached at which the acceptable distortion limit is exceeded. This distortion might be caused by microphone overload, tape saturation, or by signals that exceed the capabilities of a transformer or a transistor somewhere in the chain.

Over a half century ago some perceptive scientists at Bell Laboratories[1] made a comprehensive series of measurements of the output of many musical instruments and orchestras of several sizes. Representative peak powers they measured are listed in Table 3-1. We recall that the acoustic power involved in a normal conversational voice at a distance of 3 ft is about 50 microwatts. The bass drum measured yielded a peak power of 24.6 watts which is about 114 dB higher than the conversational voice. The 75 piece orchestra hit a peak of 66.5 watts which is 122 dB higher. The bass viol peaked out at only 78 milliwatts, the flute at 55 milliwatts. Acoustic power is a difficult parameter to measure and it is usually calculated from sound pressure measurements.

Sivian et al[1] also made some sound pressure measurements to estimate the dynamic range of an orchestra as defined by the spread between the softest and the loudest passage. A violin player was engaged and told to play at the lowest level that could be used with an audience. As measured at 3 ft distance, the average pressure in

the whole spectrum in a 15-second interval of uniform playing was 5.2×10^4 Pascals.

The highest average pressure obtained from the bass drum under comparable conditions was 1.33×10^7 Pa. On this average basis the dynamic range turns out to be

$$20 \log (1.33 \times 10^7 / 5.2 \times 10^4) = 48 \text{ dB}.$$

This is not the whole picture, however, because reproduction must go at least as low as the lowest average pressure of the violin to the highest peak of the drum which was 1.25×10^8 Pa. This increases the dynamic range to

$$20 \log (1.25 \times 10^8 / 5.2 \times 10^4) = 68 \text{ dB}$$

This illustrates the sort of reasoning behind the rather common statement that any recording or transmission system must have a dynamic range of the order of 70 dB to do justice to a symphony orchestra.

Although the dynamic range of commonly available equipment is sufficient for speech, some form of compression is necessary to reduce the 70 dB range of music to fit within the limitations of practical channels. This can be done manually by raising soft passages and reducing loud ones, or automatically by special electronic circuits. Composers, conductors, and other musically knowledgeable persons cringe at what can happen to good music in this necessary process whether it is accomplished manually or electronically.

DYNAMIC RANGE AND HOME HIGH FIDELITY

Figure 3-1 shows that portion of the ideal auditory area required for general speech and music. Figure 3-6 is more specific in regard to music and speech levels and includes common household noise levels. In a quiet living room the background noise levels restrict the softest sounds one can perceive. In a noisy living room the threshold of hearing is raised even further. Another limitation is the maximum sound levels the neighbors will endure. These household limitations further compress the net dynamic range available for lovers of reproduced music in the home.

LINE SPECTRA

The banker must know money, the doctor must know human

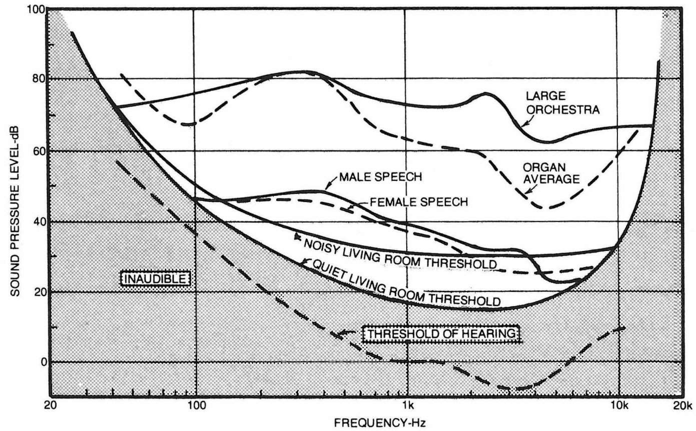

Fig. 3-6. Sound intensity variation with frequency of large orchestra, organ, and male and female speech. The threshold of hearing in average quiet and noisy living rooms severely reduces the attainable dynamic range of reproduced sounds. (Based on measurements by Sivian, Dunn, and White[1] and further work on their data by Moir[2])

physiology, and the audio worker, whether in the technical or artistic side of the business, should know the characteristics of the sounds upon which the field of audio is based. For this reason, we shall study a few types of signals and their spectra.

The ten octaves of audible sound from about 20 Hz to 20 kHz is the keyboard upon which all the sounds of human life are played. One sound is distinguished from another by different time patterns and different distributions of energy throughout the frequency range, or spectrum. Let us first direct our attention toward so-called periodic waves which repeat cycle after cycle.

The form of the simplest wave, as we have seen, is the sine wave. A "pure" sine wave is one having all its energy concentrated at a single fundamental frequency, f_o, as illustrated in Fig. 3-7(A). There are no harmonics. This is commonly referred to as a line spectrum, a term which applies whether or not there are harmonics.

A square wave of the same frequency as the sine wave is shown in Fig. 3-7(B). In this case there is also a spectral line at f_o which represents the energy of a pure sine wave of fundamental frequency, f_o, shown inscribed with a broken line within the square wave. Because the square wave is something other than a pure sine wave, harmonics are present, in this case a series of odd harmonics,

$3f_o$, $5f_o$, $7f_o$, etc., represented by lines of decreasing amplitude. In other words, if the spectrum of the square wave were searched with a narrow filter moved up and down in frequency, the spectral lines of energy of Fig. 3-7(B) would be found and their relative intensities could be measured. Every line represents a pure sine wave whose frequency can be identified in this analysis operation. Conversely, the square wave can be built up by combining pure sine waves of

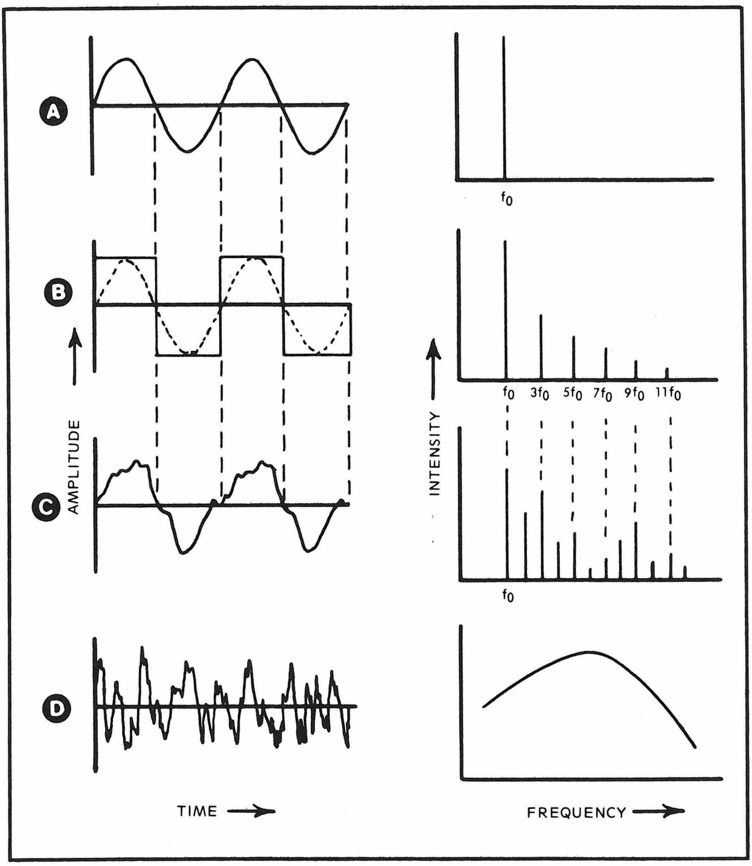

Fig. 3-7. (A) The energy of a pure sine wave is concentrated at the fundamental frequency, there are no harmonics.
(B) A symmetrical square wave of the same frequency has energy at the fundamental frequency, f_0, and also at odd multiples of the fundamental called harmonics.
(C) An irregular periodic wave of the same frequency as A, such as a violin tone, has energy at the fundamental frequency and also at both odd and even harmonics.
(D) A random noise, such as a jet aircraft sound, has no periodic structure and has a continuously distributed rather than a line spectrum.

frequencies f_o, $3f_o$, $5f_o$, $7f_o$ . . . etc. with proper amplitude and time relationships. In this sense the sine wave is considered a basic building block from which complex periodic waves of any shape can be built. Stated another way, complex periodic waves can be broken down into their sine wave components.

In Fig. 3-7(C) is a wave shape such as a violin or other musical instrument might produce. Its spectrum consists of both odd and even harmonics. In the case of the violin the relative intensities of these harmonics are determined by the construction and quality of the instrument. The greater the harmonics, the richer the tone, or timbre. The difference between the tone of a cheap violin and a Stradivarius lies principally in their different harmonic content.

CONTINUOUS SPECTRA

The periodic waves of Figs. 3-7(B) and (C) have energy concentrated at frequencies related harmonically to the fundamental frequency. The irregular, nonperiodic wave of Fig. 3-7(D) is that of the noise of a jet aircraft engine. In this case the energy is continuous through a wide band of frequencies and is called a "distributed" spectrum. It is made up of an infinite number of sine components continually shifting in amplitude and phase in a random way.

We frequently encounter combinations of continuous and line spectra. A submarine under way produces a potpourri of underwater noises including a throbbing, distributed noise produced by the propeller. Superimposed on the distributed spectrum may be one or more spectral lines associated with the whine of auxiliary machines.

HARMONICS VS. OVERTONES

There is often confusion between the musician's term "overtones" and the physicist's "harmonics". A distinction often is needed. Overtones may or may not be harmonics. The term overtone includes both higher frequency components having a harmonic relationship to the fundamental and those that do not. In other words, the *over* tones can be taken simply as *higher* tones, while harmonics are always multiples of the fundamental frequency. Some musical instruments have overtones that are not whole number multiples of the fundamental and thus cannot be classed as harmonics. Piano tones, for example, are not strictly harmonically related to the fundamental. Bells produce a wild mixture of overtones with a pitch that is somewhat indistinct because of the mixture. The overtones of drums are also not harmonically related and are thus better characterized as overtones. The non-harmonic

overtones of bells, piano and drums, however, are responsible for the rich, characteristic sounds of these instruments. Overtones of bells and piano are close enough to harmonic relationship to produce sounds of a definite pitch.

PHASE RELATIONSHIPS

Let us examine the effect of phase relationships which have been mentioned previously. In Fig. 3-8(A) a fundamental (f_o), a second harmonic ($2f_o$), and a third harmonic ($3f_o$) of the indicated amplitudes are combined to form the complex periodic wave at the top of the figure. The reader can demonstrate this to his own satisfaction by combining the instantaneous amplitudes of f_o, $2f_o$, and $3f_o$ at a given time to find the instantaneous amplitude of the resultant wave. For example, at time t_1 all the waves are passing through zero, hence a zero resultant. At time t_2, $3f_o$ is equal to zero, $2f_o$ is a positive amplitude of so many units, and f_o is somewhat larger in negative amplitude; the result is a slightly negative value for the resultant complex wave. For time t_3 the fundamental and both harmonics are positive, yielding a high positive excursion in the resultant wave.

We have an identical situation in Fig. 3-8(B) except that both $2f_o$ and $3f_o$ have been shifted to the left a quarter of their respective wavelengths. The second and third harmonics now have a different time relationship to the fundamental but all frequencies and amplitudes are the same as before. Adding ordinates of the three waves, f_o, $2f_o$, and $3f_o$, we get a resultant which has a quite different waveshape from that of Fig. 3-8(A). The difference must be attributed to the only thing that has changed: the time relationship. These relative time shifts are called "phase shifts." Phase shifts may be unintentionally introduced at many places in the communication chain, e.g., in electronic devices, in microphones, or in loudspeakers. The two resultant waves of Fig. 3-8(A) and (B) may sound quite similar to the ear in spite of the pronounced different shape because the human ear is relatively insensitive to phase differences.

Much can be learned about the nature of a complex sound signal by breaking it down into its constituent sine components. This is commonly done in the laboratory with a frequency analyzer, a specialized electronic measuring instrument capable of sorting out the harmonics of a complex periodic wave.

NOISE—THE BAD KIND

Noise is always present in all communication channels,

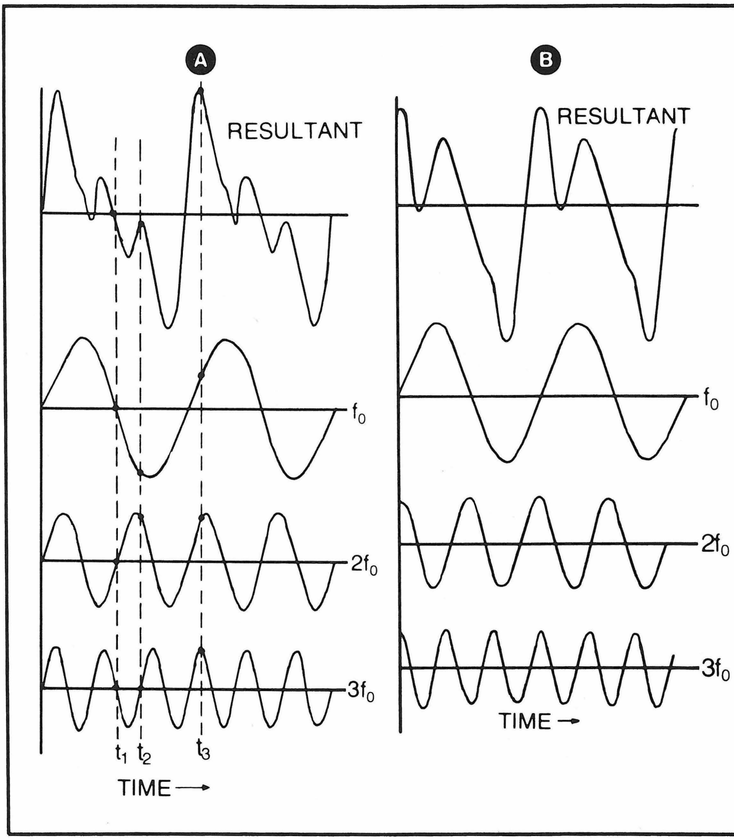

Fig. 3-8. Changing the relative phase of components of a complex periodic wave can have considerable effect on the appearance of the resultant wave. (A) A fundamental and two harmonics add to give the resultant pictured. (B) Everything is the same except that the $2f_0$ and $3f_0$ components have been shifted in phase, affecting the shape of the resultant wave. The human ear, being relatively insensitive to phase shifts, may find it difficult to detect a difference in the sound of resultants A and B.

whether it be a cozy conversation over a cup of coffee, watching an international television program arriving via satellite, or reading the morning newspaper. In this general sense "noise" may be taken to be any disturbance tending to interfere with the communication that is going on. With the conversation it might be the clatter of dishes from the kitchen, or snow on the television screen, or the intrusion of too many advertisements in the newspaper. This is "bad" noise comparable to the noise we have discussed in connection with dynamic range.

NOISE—THE GOOD KIND

A good kind of noise? Defining noise as unwanted sound fits the system noise considered previously, but noise is becoming an increasingly important tool for measurements in acoustics as we shall see in Chapter 14. The noise is not necessarily different from the bad noise interfering with our listening to a favorite recording, it is just that the noise is put to a specific use.

In acoustical measurements the use of pure tones might be very difficult to handle while a narrow band of noise centered on the same frequency would make satisfactory measurements possible. For example, if a studio is filled with a pure tone signal of 1000 Hz from a loudspeaker, a microphone picking up this sound will have an output that varies greatly from position to position due to room resonances. If, however, a band of noise one octave wide centered at 1000 Hz were radiated from the same loudspeaker, the level from position to position would tend to be more uniform, yet the measurement would contain information on what is happening in the region of 1000 Hz. Such measuring techniques make sense as we are usually interested in how a studio or listening room reacts to the very complex sounds we are recording or reproducing, rather than to steady, pure tones.

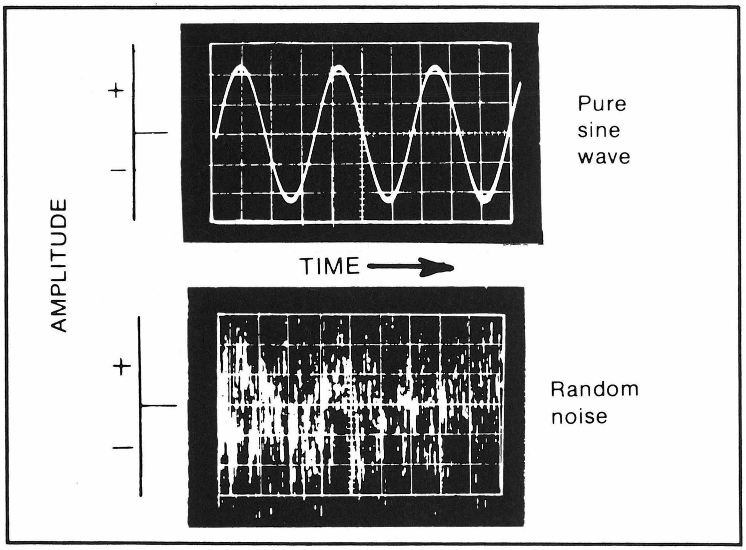

Fig. 3-9. Cathode ray oscillograms of a pure sine wave and of random noise. Random noise may be considered made up of sine waves which are continually shifting in amplitude, phase, and frequency.

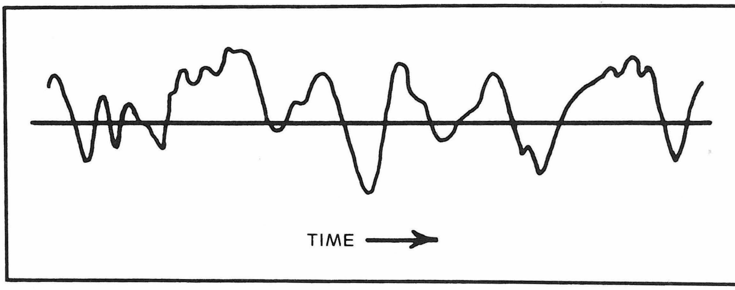

Fig. 3-10. A section of the random noise signal of Fig. 3-9 spread out in time. The non-periodic nature of a noise signal is evident, the fluctuations are random.

RANDOM NOISE

Random noise is generated in any electrical circuit and minimizing its effect often becomes a very difficult problem. Heavy ions falling back on the cathode of a thermionic vacuum tube produce noise of a relatively high amplitude and wide spectrum and the introduction of some gas molecules into the evacuated space will produce even more noise. Today a random noise generator is made with a silicon diode or other solid-state device followed by an amplifier, voltmeter, and attenuator.

In Fig. 3-9 a pure sine wave and a random noise signal are

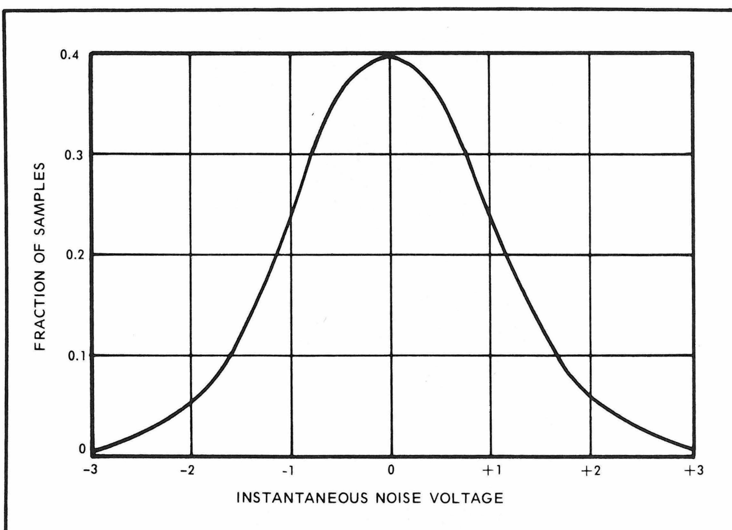

Fig. 3-11. The proof of randomness of a noise signal lies in the sampling of instantaneous voltage, say, at 1000 points equally spaced in time and plotting the results. The familiar bell-shaped gaussian distribution curve results if the noise is truly random.

compared as viewed on a cathode ray oscilloscope. The regularity of the one is in stark contrast to the randomness of the other. If the horizontal sweep of the oscilloscope is expanded sufficiently and a snapshot is taken of the random noise signal it would appear as in Fig. 3-10.

Noise is said to be purely random in character if it has a "normal" or "Gaussian" distribution of amplitudes.[3] This simply means that if we sampled the instantaneous voltage, say, at a thousand equally spaced times, some readings would be positive, some negative, some greater, some smaller, and a plot of these samples would approach the familiar Gaussian distribution curve of Fig. 3-11.

WHITE AND PINK NOISE

The energy of such random noise is spread more or less uniformly over a wide frequency range. Noise which has uniform energy per hertz can be analyzed with filters of various bandwidths. The wider the bandwidth, the higher the reading. Thus, the shape of

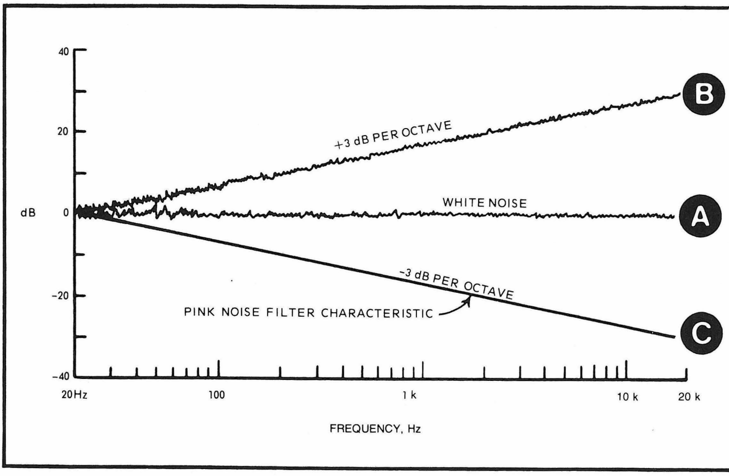

Fig. 3-12. Random noise has constant energy per Hz. If the spectrum of random noise is measured with a wave analyzer of fixed bandwidth, the resulting spectrum will be flat with frequency as A. If measured with an analyzer whose passband width is a given percentage of the frequency to which it is tuned, the spectrum will slope upward at 3 dB per octave, as in B. By processing the white noise spectrum of A with a filter that slopes downward at 3 dB per octave, such as C, a flat response results when constant percentage bandwidth filters are used such as octave or one-third octave filters. In measuring a system pink noise is applied to the input and, if the system is flat, the read out response will be flat if one-third octave filters, for example, are used.

the measured noise spectrum is determined by the type of analyzer used. If the noise is analyzed with a filter that has a fixed bandwidth that can be tuned to any frequency in the audible spectrum, we find the analyzer output to be essentially the same no matter to what frequency it is tuned. In other words, the noise spectrum would be as shown in Fig. 3-12(A). If the random noise signal is analyzed by an analyzer which has a filter whose passband width is a given percentage of the frequency to which it is tuned, the analyzer output would increase with frequency. In Fig. 3-12(B) the same noise has been analyzed with a tunable filter one-third octave wide. At 100 Hz the bandwidth is only 23 Hz but at 10 kHz the bandwidth is 2300 Hz. Obviously there is much greater noise energy in a one-third octave band centered at 10 kHz than one centered on 100 Hz. This means that analyzing the same noise signal with a one-third octave analyzer yields a spectrum which rises with frequency at a rate of 3 dB per octave.

As the one-third octave system is so convenient and so widely used in acoustical measurements, it is customary to process the noise of Fig. 3-12(B) by passing it through a pink noise filter having a minus 3 dB per octave slope as in Fig. 3-12(C) which brings graph B down to the white noise characteristic of graph A and thus well adapted to many acoustical and other measurements. If the system being measured is flat, then using a pink noise signal with a one-third octave real time analyzer yields a flat response read-out. For this reason pink noise sources are widely used because one-third octave analyzers (and other constant percentage analyzers) are widely used.

The application of color terms to noise may sound quaint, but it comes about quite naturally. White light is a mixture of all colors. If white light is analyzed with optical filters, various colors result. The low frequency (long wavelength) end of the visible spectrum is red, the high frequency end is violet. Thus "white" noise has a continuous distribution of energy like white light. Noise energy distributed throughout the audible spectrum, but with the low frequency ("red") end emphasized, is called, very logically, "pink noise". Extending this idea a bit, the noise of Fig. 3-12(B) can then, just as logically, be called "violet noise" (or lavender? orchid? mauve?).

FORMS OF DISTORTION

Our discussion of the various signals encountered in audio is incomplete without at least an acknowledgment of what can happen

to the precious signal in passing through transducers, amplifiers, and various forms of signal processing gear. Here is an incomplete list:[4, 5, 6]

Bandwidth Limitation. If the passband of an amplifier cuts lows or highs, the signal output is different from the input. If the scratch filter reduces record surface noise, the overall effect may be improved but basically the signal itself is the poorer for it.

Nonuniform Response. Peaks and valleys within the passband also alter the signal waveshape.

Distortions in Time. If tape travels across the head at any other than the recording speed, the frequency components are shifted up or down in frequency. If there are slow or fast fluctuations in that speed, wow and flutter are introduced and the signal is degraded.

Phase Distortion. Any phase shifts introduced upset the time relationship between signal components.

Dynamic Distortion. A compressor or expander changing the original dynamic range of a signal is a form of distortion.

Crossover Distortion. In class-B amplifiers, in which the output devices conduct for only half of the cycle, any discontinuities near zero output result in what is called "crossover" distortion.

Nonlinear Distortion. If an amplifier, for example, is truly linear, there is a one-to-one relationship between input and output. Feedback helps to control nonlinear tendencies. The human ear is not linear. When a pure tone is impressed on the ear, harmonics can be heard. If two loud tones are presented simultaneously, sum and difference tones are generated in the ear itself; and these tones can be heard as can their harmonics. A cross-modulation test on an amplifier does essentially the same thing. If the amplifier (or the ear) were perfectly linear, no sum or difference tones or harmonics would be generated. The production within the black boxes of frequency components that were not present in the input signal is the result of nonlinear distortion.

Transient Distortion. Strike a bell and it rings. Apply a steep wavefront signal to an amplifier and it may ring a bit too. For this reason, signals such as piano notes are difficult to reproduce. Tone burst test signals are an attempt to explore the transient response characteristics of equipment, as are square waves. Transient intermodulation (TIM) distortion, slew induced distortion, and other sophisticated measuring techniques have been devised to evaluate transient forms of distortion in systems.

When one considers all these terrible things that can happen to precious signals, one could become despondent; either that or jarred awake by the challenge of circumventing all these potential pitfalls! We hope it will be the latter for the following chapters reveal other cruel acoustical fates waiting to pounce on unsuspecting signals.

HARMONIC DISTORTION

The harmonic distortion method of evaluating the effects of circuit nonlinearities is probably the oldest and the most universally accepted method. It certainly is the easiest to understand. In this method the device under test is driven with a sine wave of high purity. If the signal encounters any nonlinearity, the output waveshape is changed, i.e., harmonic components appear that were not in the pure sine wave. A special analysis of the output signal is made to measure these harmonic distortion products. The most revealing method is to use a wave analyzer having a constant passband width of, say, 5 Hz, which can be swept through the audio spectrum. In Fig. 3-13 are illustrated results of such a measure-

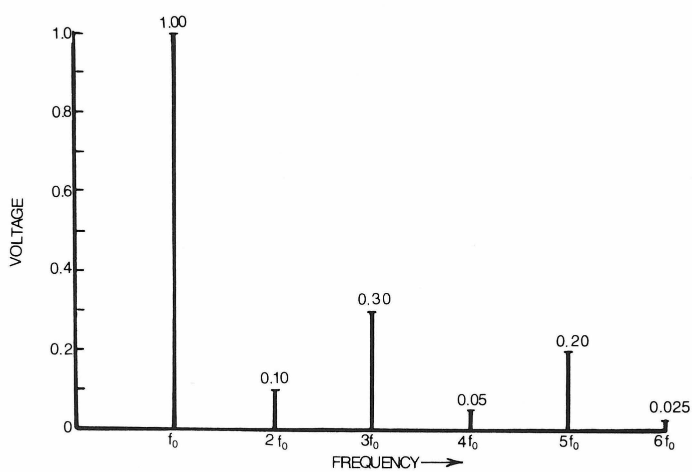

Fig. 3-13. A distorted periodic wave is measured with a constant bandwidth wave analyzer. The fundamental, f_o, is set for some reference voltage, taken here as 1.00 volt. Tuning the wave analyzer to $2f_o$ the second harmonic amplitude is measured as 0.10 volt. The wave analyzer is tuned successively to $3f_o$, $4f_o$ and other harmonics yielding amplitudes of each harmonic as shown. The root-mean-square of the harmonic voltages is then compared to the 1.00 volt fundamental to find the total harmonic distortion expressed in percentage.

Table 3-2. Harmonic Distortion Products.

Fundamental f_0 = 1 kHz, 1.00 volt amplitude

Harmonic	Volts		$(\text{Volts})^2$
2nd Harmonic $2f_0$	0.10		0.01
3rd Harmonic $3f_0$	0.30		0.09
4th Harmonic $4f_0$	0.05		0.0025
5th Harmonic $5f_0$	0.20		0.04
6th Harmonic $6f_0$	0.025		0.000625
7th and higher	(negligible)		
		Sum	0.143125

ment. The wave analyzer is first tuned to the fundamental, f_0 = 1 kHz, and the level is set for a convenient 1.00 volt. The wave analyzer is then tuned to the 2 kHz region and adjusted until the $2f_0$ second harmonic is found. The voltmeter, which is a part of the analyzer, reads 0.10 volt. The third harmonic at 3 kHz gives a reading of 0.30 volt, the fourth a reading of 0.05 volt and so on up the frequency scale. Beyond $6f_0 = 6$ kHz no measurable components were found after diligent search. The data are then assembled in Table 3-2. The total harmonic distortion (THD) may then be found from the expression:

$$\text{THD} = \frac{\sqrt{(e_2)^2 + (e_3)^2 + (e_4)^2 \ldots (e_n)^2}}{e_o} \times 100 \qquad \textbf{(3-1)}$$

where $e_2, e_3, e_4 \ldots e_n$ = voltages of 2nd, 3rd, 4th, etc. harmonics
e_o = voltage of fundamental

In Table 3-2 the harmonic voltages have been squared and added together reducing Eq. (3-1) to:

$$\text{THD} = \frac{\sqrt{0.143125}}{1.00} \times 100$$
$$= 37.8\%$$

A total harmonic distortion of 37.8% is a very high distortion which would make any amplifier sound horrible on any type of signal, but the example has served our purpose in illustrating just what THD is and one method for obtaining it.

Wave analyzers are expensive, high precision instruments

which are rarely found in equipment service shops. A very simple adaptation of the THD method is, however, widely used. Consider Fig. 3-13 again. If the f_o fundamental were adjusted to some known value and then a notch filter were adjusted to f_o essentially eliminating it, only the harmonics would be left. Measuring these harmonics all lumped together with an RMS (root mean square) meter, we really accomplish what was done in the square root portion of Eq. (3-1). Comparison of this RMS measured value of the harmonic components with that of the fundamental and expressing it as a percentage gives the total harmonic distortion.

In Fig. 3-14 an undistorted sine wave is sent through an amplifier which clips positive peaks. On the left the flattening of the positive peaks with 5% THD is evident and below is what the combined total of all the harmonic products look like with the fun-

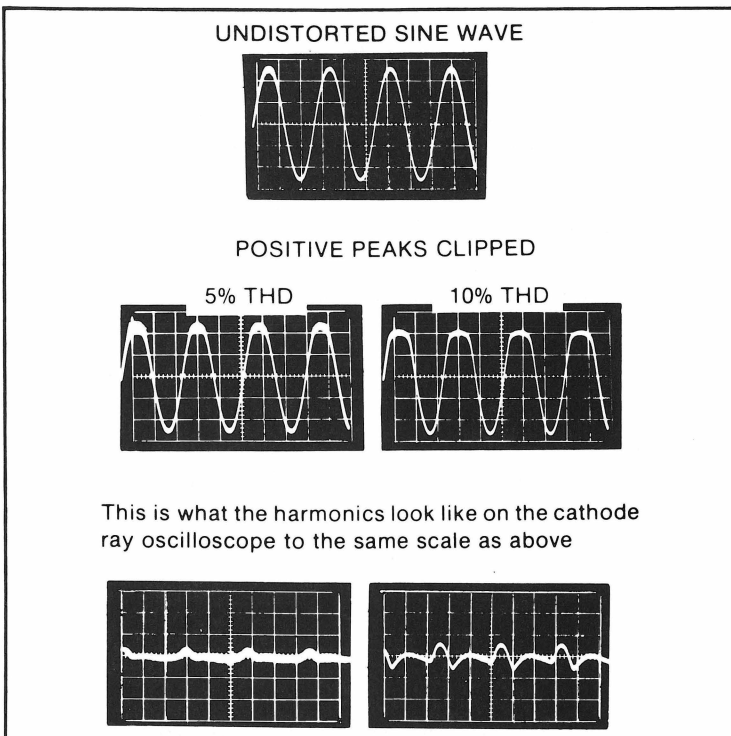

Fig. 3-14. Cathode ray oscillograms show pure, undistorted sine wave which is applied to the input of an amplifier which clips the positive peaks of the signal. The appearance of the clipped sine wave for 5% and 10% total harmonic distortion is shown. If the fundamental is rejected by a notch filter, the summed harmonics appear as shown.

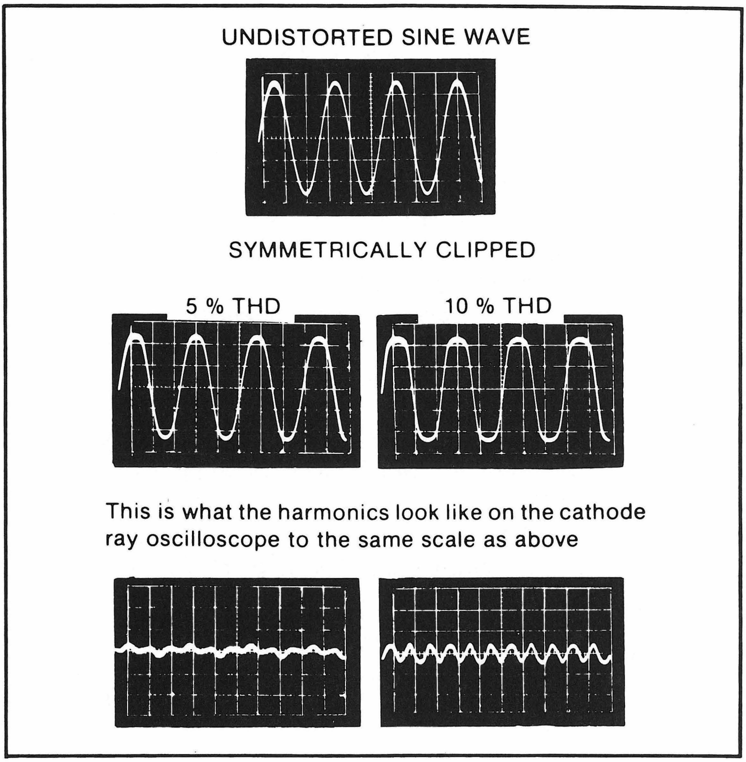

Fig. 3-15. Cathode ray oscillograms show a pure, undistorted sine wave which is applied to the input of an amplifier which clips both positive and negative peaks in a symmetrical fashion. The appearance of the clipped sine wave is shown for 5% and 10% total harmonic distortion. The appearance of the harmonics alone, with the fundamental filtered out, is also shown.

damental rejected. On the right is shown the effect of greater clipping to yield 10% THD. In Fig. 3-15 is shown what happens when the sine wave passing through the amplifier is symmetrically clipped on both positive and negative peaks. The combined distortion products for symmetrical clipping have a somewhat different appearance, but they measure the same 5% and 10% THD.

In all of this exercise we must remember that consumer type power amplifiers commonly have specifications listing total harmonic distortion nearer 0.05% rather than 5% or 10%. In a series of double-blind subjective tests Clark[7] found that 3% distortion was audible on different types of sounds. With carefully selected material (such as a flute solo) detecting distortions down to 2% or 1% might be possible. A distortion of 1% with sine waves is readily audible.

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4

Human Hearing

Directly or indirectly, all questions connected with (sound) must come for decision to the ear . . . . . and from it there can be no appeal.
Lord Rayleigh

Each person interested in high fidelity recording and reproduction of sound knows how important human hearing is. Everything revolves around it. "The specifications look good, how does it sound?" No matter what the measurements say, the ear is the court of last appeal, the final arbiter. As long as you are buying or building to suit only yourself the problems are minimized—you have only yourself to please. The problems arise when it becomes necessary for others to be brought into agreement. People are different. They are different in physical and pyschological attributes, different in cultural upbringing, different in tastes.

PSYCHOACOUSTICS

In the opening pages of Chapter 1 the relationship between stimulus and sensation, between meter readings and human reaction, was discussed. This distinction was clearly developed in the first edition of this book, but during the intervening decade a renewed and wholesome emphasis on the human reaction part has appeared. The buzzword is now *psychoacoustics*. The accurate and dependable evaluation of sound quality is now a tremendously important part of the audio industry and the steady improvement in amplifiers, loudspeakers, and other components requires increas-

ingly keener distinctions. The general awakening to the psychoacoustical facts of life and the growing number of tests involving panels of listeners bodes well for the future.

HEARING VS. LISTENING

At first glance, hearing and listening might be considered synonymous. Although both involve the hearing mechanism, there is an important distinction. The word hearing simply means becoming aware of sounds through stimulation of the auditory nerves. The sounds of everyday life that incessantly beat upon our eardrums contribute to our sense of aliveness, but our reactions to them are largely subconscious. Listening, on the other hand, involves a conscious effort, a focusing of attention on the content of the stimulus. Listening is the very heart of activity in audio, both for pleasure and for technical analysis.

EARS—GOLDEN AND TIN

There is surely such a thing as personal taste in music. Some prefer Beethoven, some Shankar. Assemble all the Beethoven lovers and one would find that some like the knobs set one way and some another. We would not want to have sterile agreement and conformity in such things—each person is a unique creation and differences between individuals add zest and interest to our lives (*Vive le difference!*).

There is an important point here that goes beyond esthetic interests and cultural conditioning. Maybe listener A set the knobs the way he did because his hearing is deficient in the highs! Measurements of hearing acuity have been made upon hundreds of thousands of people and it has been determined that "normal" hearing embraces a considerable spread. It is still very useful to study the characteristics of that "average" ear which, in a sense, describes most of us but which, in another sense, doesn't quite fit any of us.

Listener B may have adjusted the knobs differently because he is a professional who has spent his life not just listening, but listening critically and analytically. Perhaps he rolled off the lows a bit to minimize an irritating room mode, or reduced the volume to control some distortion which Listener A and others didn't even hear. Although the phrase is used sometimes in derision, saying that Listener B has "golden ears" would make sense because of his uncanny ability, developed over a decade or two of critical listening, to detect flaws. There may be as much difference between the

ability of the man with the golden ears and the average audio enthusiast as there is between the audio enthusiast and the youth walking along the beach absorbed in the raucous output of a stereo player perched on his shoulder.

Critical and analytical listening can add a tremendous new dimension to one's enjoyment of music, but we should not stop there. If we do, we lose touch with the music itself. This is somewhat like the lighthouse enveloped in pea soup fog. After several days and nights of incessant foghorn blasts, the horn went dead. The lighthouse keeper suddenly looked up at his wife and said, "Sh-h-h, what was that?"

TRAINING FOR LISTENING

The epitome of listening skill would seem to be that of the sound mixer who has worked professionally for 10 or 20 years in the critical evaluation of sound. The ability of such a person to detect seemingly inaudible distortions, colorations, hums, and other noises never fails to amaze the casual observer. How did this person attain such an enviable skill? The listening experiences came on a random basis over long stretches of time. Would it be possible to present to a neophyte similar listening experiences on a carefully programmed basis and thus accelerate the acquiring of such a skill? The author has made a pioneering effort in this direction [1, 2] and it would seem that the prospects for such training in critical listening are very bright. Such organized training, of course, will never replace the need for experience, but it promises to accelerate the acquiring of listening skill.

HOW WE HEAR

While the beauty of our external ears may be nothing to brag about, the hearing mechanism of which they are a part is truly a wondrous system. Capable of hearing, under proper conditions, the infinitely delicate tatto of air molecules on the eardrum, the ear can handle, without damage, sounds millions of times stronger. From the sighing of leaves in the wind to the roar of a jet engine, the ear can tell one sound from another in an intricate way not yet fully comprehended.

AURAL MECHANISM

The external ear helps collect the sound and send it down the auditory canal to the eardrum (Fig. 4-1). The vibrations of this

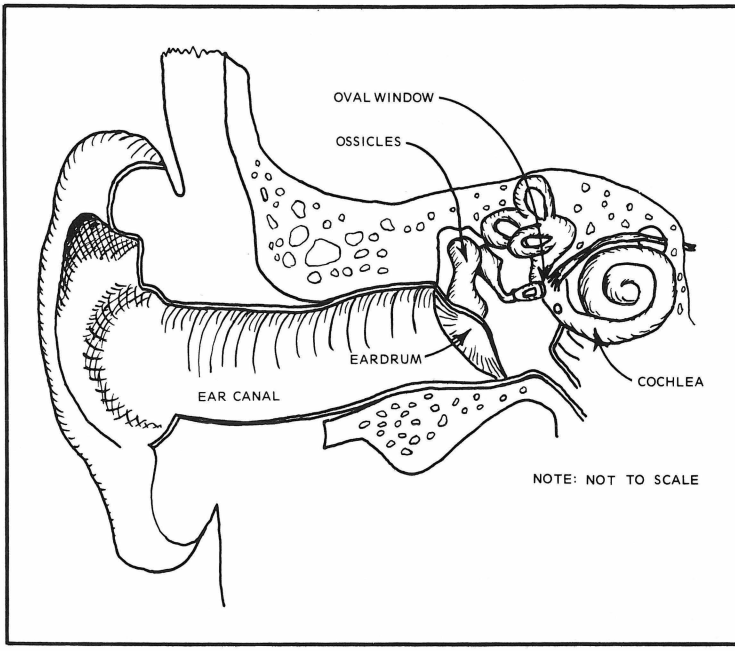

Fig. 4-1. Cross section of the human ear. The airborne sound stimulus causes the eardrum to move. This mechanical movement is transmitted to the fluid within the cochlea by the ossicles. Sensitive hair cells in the cochlea are connected to the brain through the auditory nerve. Amplification of the stimulus takes place in the ear canal and optimum design of the mechanism of the middle ear assures efficient transfer of the stimulus to the inner ear.

diaphragm of thin skin are transmitted across the air cavity of the middle ear by a linkage of three delicate bones, the ossicles, often called the hammer, the anvil, and the stirrup because of their shape. Movements of these bones are transmitted to the liquid inside the cochlea through the oval window opening into the bony case of the cochlea. Within the cochlea is the sensitive Organ of Corti with its hair cells connected to the auditory nerve. These hair cells are stimulated in orderly patterns by sound traveling in the fluid. The entire cochlea is no larger than the tip of one's little finger and there is about one drop of fluid within, surely a marvel of miniaturization.

AUDITORY NERVE

The stimulation of the sensory hair cells generates an electrical trigger potential which, in turn, initiates neural impulses in the auditory nerve. This nerve is composed of thousands of nerve fibers, each of which is a chain of nerve cells or neurons surrounded

by an insulating sheath. The signals carried by the fibers, however, are nothing like the direct or alternating currents we normally associate with electrical conductors, but rather an on-off, all or none action. Through electrochemical reaction, pulses travel from neuron to neuron at speeds between 3 and 300 ft per second, the larger the nerve, the faster the propagation. The number of firings in a given time interval increases as the stimulating intensity rises. There is a complex, little understood coding process involved by which differences in frequency, loudness, etc. are processed and perceived.

ANALYSIS OF EAR FUNCTION

To study the operation of the human ear it is well to abandon the physiological representation of Fig. 4-1 and move on to the functional representation of Fig. 4-2. This move is made with trepidation and apology because the crudeness of Fig. 4-2 is an afront to the beautiful, efficient, and highly miniaturized nature of the ear. Yet, if we can convey something of the ear's amazingly purposeful design, the lack of refinement may be forgiven.

Because the sounds picked up by the ear are so feeble, from an energy standpoint, they must be handled efficiently in order to reach the cochlea, the true sensing part of the ear, with sufficient magnitude to be detected. Referring to Fig. 4-2, we first observe the outer ear which involves the pinna, the external flap which serves as a collector of sound, especially at the higher frequencies. The degree of this effect can be sensed by cupping the ear with a hand which increases the collecting area somewhat. Recent dis-

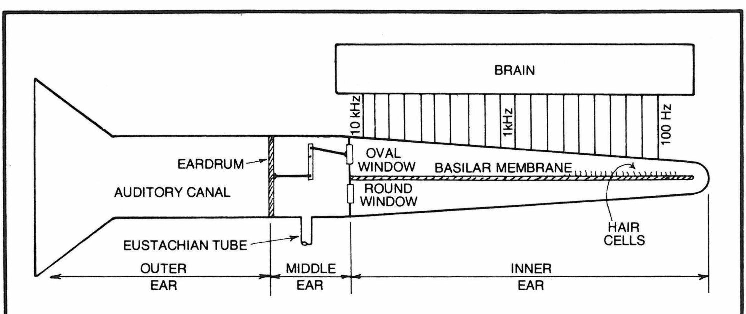

Fig. 4-2. A crude representation of the human ear to illustrate the function of the outer, middle, and inner ear. The vibration of the eardrum is transmitted to the oval window of the cochlea by three tiny bones, the ossicles. Traveling waves set up in the fluid of the cochlea stimulate the hair cells, initiating nerve impulses which travel to the brain.

coveries suggest that this collection effect may be one of the minor functions of the pinna. The convolutions of the pinna may not be simple adornments, but producers of comb filter reflections giving cues as to direction of arrival of sounds.

The auditory canal opens into the pinna on one end and is terminated by the eardrum on the other. This tube turns out to be a ¼ wavelength resonant system at midband frequencies, much as an organ pipe closed at one end. Purely from this resonant tube effect, sound pressure falling on the open end is amplified about 10 dB at the eardrum in the 3-4 kHz region. There is another amplifying action at work. A plane progressive sound wave striking a person from the front is diffracted by the head. This head diffraction increases the sound pressure at the eardrum by another 10 dB. The combination of the ¼ wave resonance effect and the head diffraction effect results in an amplification of about 20 dB in the very important mid-frequency range of the audible spectrum. This, of course, is in the direction of increased sensitivity of the ear at these frequencies.

MIDDLE EAR

The middle ear is an acoustically isolated, air-filled cavity. It is connected to the outside world only through the Eustachian tube. This tube has a high acoustic impedance and, as a result, sound waves falling on the ear are kept from acting on the middle ear side of the eardrum, neutralizing its effect. It does allow equalization of static air pressure on both sides of the eardrum, however. Clogging this tube by infection or a cold, or changing altitude in airplanes, can create discomfort when pressure is not the same on both sides of the eardrum. In terms the audio person is familiar with, the eardrum acts as an acoustic suspension device, working against the compliance of the air trapped in the middle ear.

In addition to the eardrum and the Eustachian tube, the cochlea (inner ear) interfaces the middle ear in the form of two tiny membrane diaphragms, the oval window and the round window. The round window serves as a pressure release for the fluid filling the cochlea. Three tiny bones, the ossicles, form a mechanical bridge between the eardrum and the oval window. Popularly termed the hammer, anvil, and stirrup because of their shapes, these bones constitute an amazing linkage giving a mechanical advantage of approximately 3:1. This is a quite linear action until loud sounds result in excessively high amplitudes. At this point a limiting action sets in which serves to protect the ear from damage.

The eardrum has an area of about 80 sq. mm, and that of the

oval window attached to the foot of the stirrup is about 3 sq. mm a ratio of areas of about 27. With the force advantage of 3 from the lever action of the three ear bones added to this area ratio of about 27, an overall force increase of about $3 \times 27 = 81$ results. The obvious purpose of all this mechanism is to match the acoustic impedance of air with that of the fluid of the inner ear which is some 3600 times greater. To accomplish this impedance match requires a pressure or force ratio of $\sqrt{3600} = 60$ which is reasonably close to the 81 above. When the ossicle mechanical advantage, taken above as 3, is taken as the more descriptive 1.3 to 3, the impedance match is near perfect. It is interesting to note that the ossicles of a baby are fully formed and do not grow appreciably as the baby grows. To do so would upset the efficiency of sound transfer from the eardrum to the fluid of the cochlea. It is difficult to avoid the conclusion that beneficient design has framed such awesome acoustical amplifiers and mechanical impedance transformers.

INNER EAR

We have discussed only the acoustical amplifiers and the mechanical impedance matching devices of the outer and middle ear. These are relatively well understood. The intricate operation of the cochlea and the brain to which it is attached, however, is very much clouded in mystery but extensive research is steadily adding to our knowledge. The cochlea as a sensory organ must be considered as an extension of the brain and share with the brain a complexity yet to be revealed with full understanding.

The portrayal of the middle and outer ear of Fig. 4-2 shows them as models of precision compared to the crudity employed to suggest the functions of the inner ear. Here the tiny snail-shaped cochlea has been unrolled and its 32 mm length magnified to see some of the detail of its structure. As stated, the cochlea is filled with a fluid, about a drop of it. The basilar membrane and another membrane called Reissner's membrane (not shown) divides the elongated cavity into three compartments along the length of the cochlea. The basilar membrane supports the hair cells which convert fluid motion into nerve impulses. The vibration of the oval window, set in action by movement of the stirrup, transmits sound vibrations to the fluid of the cochlea. Waves are set up which travel toward the distant end. The high frequencies travel only a short distance because of their great attenuation, the low frequencies travel farther.

Helmholtz attributed frequency selective properties to the

cochlea in the latter part of the nineteenth century. He envisioned the cochlea as a series of tuned strings, like a piano. Békésy [3,4], however, found traveling waves. The electrical engineers (Békésy was a telephone engineer early in his career) would consider the basilar membrane a nonuniform (dissipative) transmission line as the high frequencies travel a short distance and low frequencies as far as 32 mm to the distant end. A sort of resonance effect is set up with the envelopes of the traveling waves resulting in maximum amplitude of vibration of the basilar membrane at different positions for different frequencies, high frequencies near the oval window, low frequencies at the opposite end. There is uncertainty as to whether this mechanical filtering action alone can explain the impressive selective ability of the ear disclosed by other experiments. For example, at 1 kHz, subjects can detect a difference in frequency as small as 3 Hz. There are some 25,000 hair cells and the observed "tuning curves" resulting from masking experiments suggest the possibility that each hair cell acts like a bandpass filter. The sharpness of frequency selectivity of the ear is shown in Fig. 4-3 compared to that of a third-order ⅓ octave filter.[5]

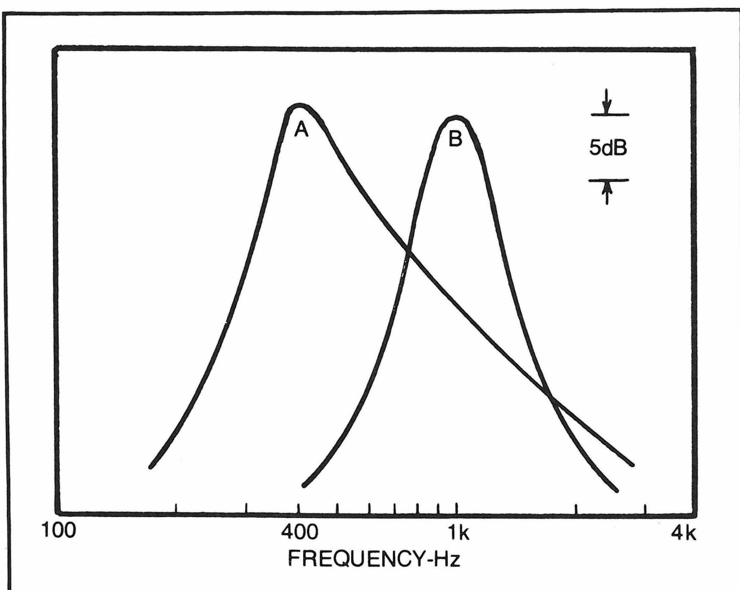

Fig. 4-3. Comparison of the frequency selectivity of the human ear and that of a ⅓ octave filter: (A) considering a hair cell as a bandpass filter at 400 Hz as determined by masking experiments, (B) response of a third order ⅓ octave filter commonly used in acoustical measurements.

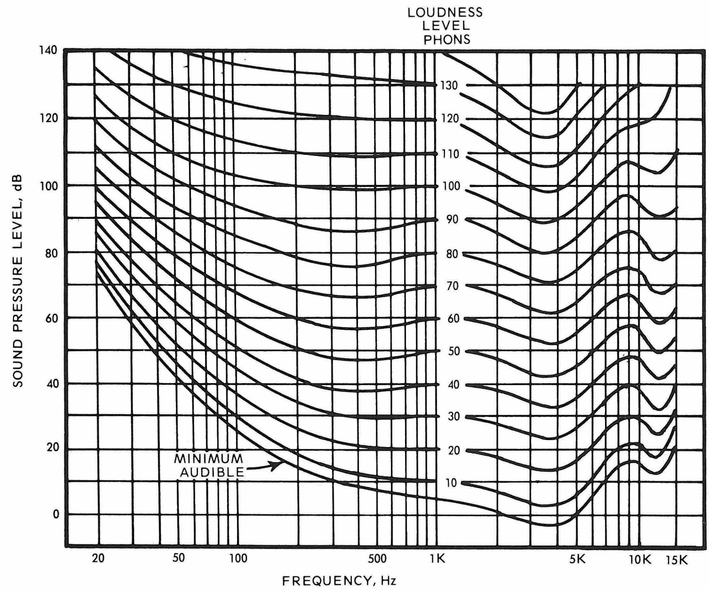

Fig. 4-4. Equal loudness contours of the average human ear, commonly called the Fletcher-Munson curves. The 70-phon contour shows the levels of sound required at each frequency to make it sound as loud as the 70 dB tone at 1 kHz. These show the lack of flat response of the ear.

EQUAL LOUDNESS CONTOURS

In Chapter 1 it was suggested that there is no one-to-one relationship between sound intensity, a physical quantity, and loudness perceived by the human hearing apparatus. Figure 4-4 is a refinement of the famous Fletcher-Munson equal-loudness contours which show that loudness and sound pressure levels are related in a complex, contorted way. These contours result from measurements on many people and may be taken as characteristic of average hearing in humans. But how are the curves obtained?

Figure 4-4 displays a family of equal-loudness contours of the human ear responding to pure tones at many loudness levels. The loudness level of 10 phons is arbitrarily made to correspond to a sound pressure level of 10 dB at 1 kHz. This is the definition of the phon. A 100 Hz tone must have a sound pressure level 20 dB higher, or 30 dB, to be judged to have the same loudness as the 1 kHz tone. The 10-phon loudness contour is located by plotting the sound pressure level required experimentally to make tones of various

frequencies as loud as the 1 kHz tone. The 20 phon loudness contour and the others are built up in the same way.

The family of equal loudness contours of Fig. 4-4 is not the last word on the subject of loudness. In fact, the phon is a physical, not a psychoacoustical term. The two systems have been tied together, however, by another arbitrary definition such as that defining the phon. A loudness (differentiating from loudness level) of 1 sone is arbitrarily defined as the loudness of a 1 kHz tone at 40 phons. This gives us point #1 in Fig. 4-5. People agree quite well when asked to

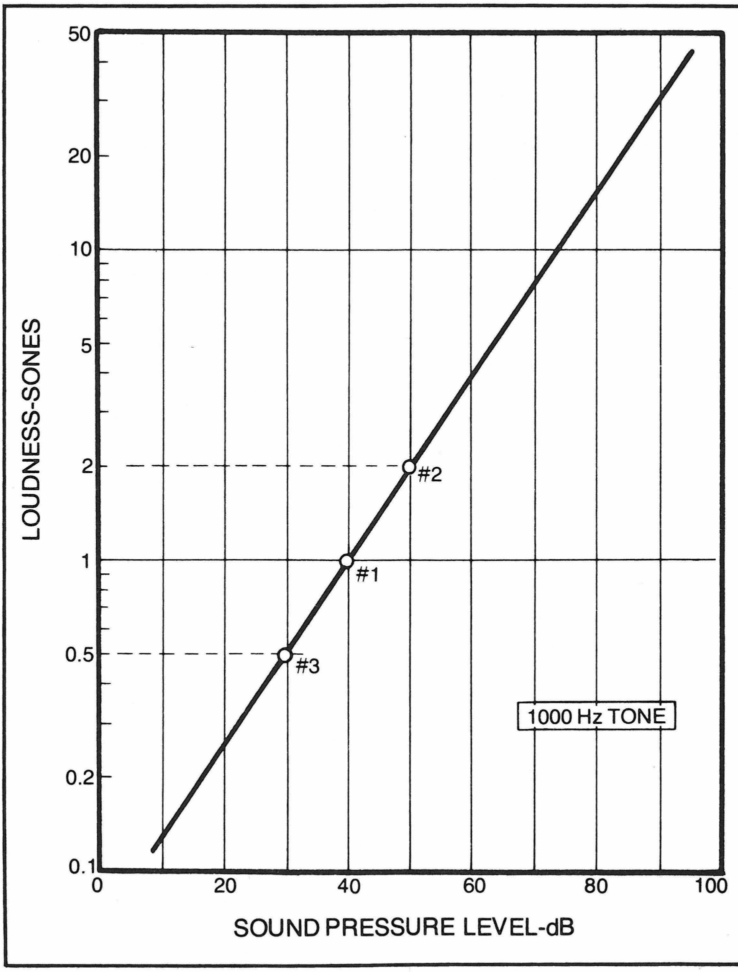

Fig. 4-5. This graph illustrates the relationship of sound pressure level to loudness in sones. A loudness of 1 sone is arbitrarily taken as a loudness level of 40 phons. This graph applies only to 1 kHz tones.

judge a sound twice as loud or half as loud as another. Starting with the 1 kHz tone of 1 sone loudness, subjects were asked to judge a sound twice as loud. This loudness is 2 sones and is plotted in Fig. 4-5 as point #2. Repeating for a 1 kHz tone half as loud as the 1 sone tone yields point #3 in Fig. 4-5. Other tests show that this is a straight line when the sones are plotted on a logarithmic scale. Now, for 1 kHz tones, we have the relationship between subjective loudness and the physical measurement of sound pressure level. The estimates of doubling and halving of loudness have been associated with 10 dB change in sound pressure level. Some have thought that this should be 6 dB instead of 10 dB and later thought suggests that 9 dB is the proper interval. The nice thing about sones is that loudness in sones may be added to give a combined loudness in sones which can then be converted to loudness level in phons or sound pressure level.

LIMITS OF HEARING

No sounds are audible to the average human ear if they have

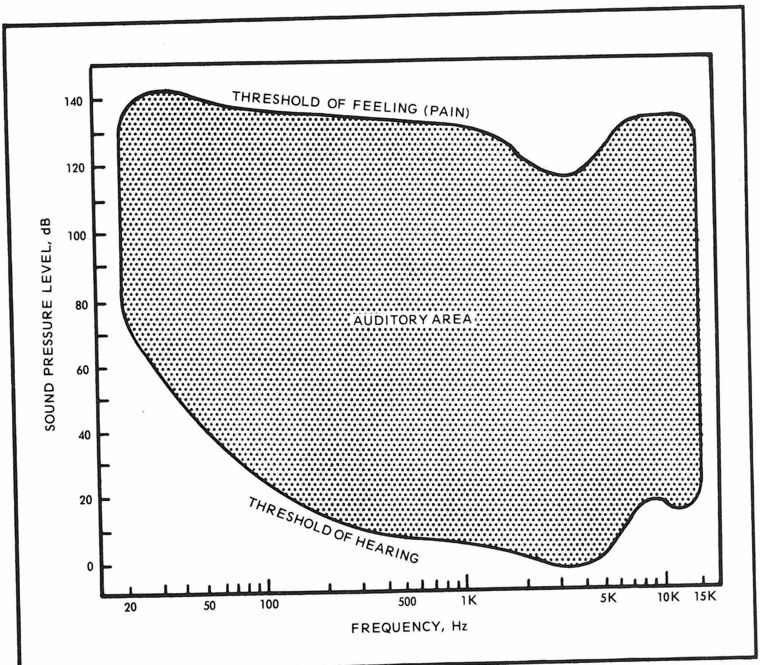

Fig. 4-6. All the sounds of life that we can perceive must fall within the shaded area. It is bounded at low levels by the threshold of hearing, at high levels by the threshold of pain, and by the low and high frequency limits of hearing.

sound pressure levels less than the "minimum audible" curve of Fig. 4-4. This represents the threshold of hearing. If any particular tone is gradually increased in level, a point is reached at which the sensation of sound gives way first to a tickling sensation and then, with further increase, to a sensation of pain. This occurs at about the 130 phon curve of Fig. 4-4.

The "auditory area" of Fig. 4-6 is bounded below by the threshold of hearing (or minimum audible) and above by the threshold of feeling or pain. All the tones of every frequency and intensity to which the ear is capable of responding are contained within this area.

FREQUENCY RESPONSE OF THE EAR

We can strain hard for flat response of amplifiers, microphones, and loudspeakers, but the stark fact is that the frequency response of the normal human ear is not flat. In fact, the ear's frequency response changes with loudness of the sound, a condition that would wreak havoc in the electronic area! A knowledge of this characteristic of the ear is of great importance to all who are called upon to judge sound quality.

Each contour of Fig. 4-4 represents the sound pressure level required to make tones of different frequencies sound as loud as the corresponding 1 kHz tone. Looking carefully at the 60 phon contour we see that at 20 Hz the sound pressure level must be 100 dB instead of 60 dB as at 1 kHz. This means that the ear is 40 dB less sensitive at 20 Hz than at 1 kHz. Similarly, at 100 Hz it is 8 dB less sensitive than at 1 kHz. From the data on the 60 phon contour we can plot the frequency response of the average human ear at the 60 phon loudness level. This has been done in Fig. 4-7. A similar curve has been plotted for the 80 phon loudness level for comparison.

The audiophile may be somewhat taken aback to learn that the ear itself is far from "flat, 20 to 20,000 Hz" which would be the 0 dB line in Fig. 4-7. Why worry about flatness of response of amplifiers and loudspeakers if the ear is like this? Just what should be our goal in our striving for perfection in reproduction and transmission of sound?

Let us go to the concert hall, sit in the best seat, and listen to a symphony orchestra. We hear the music and enjoy it very much, not particularly conscious of problems of dynamic range, frequency response, loudness level, etc. Perhaps our goal should be to reproduce in our home the music as heard in the concert hall as perfectly as possible.

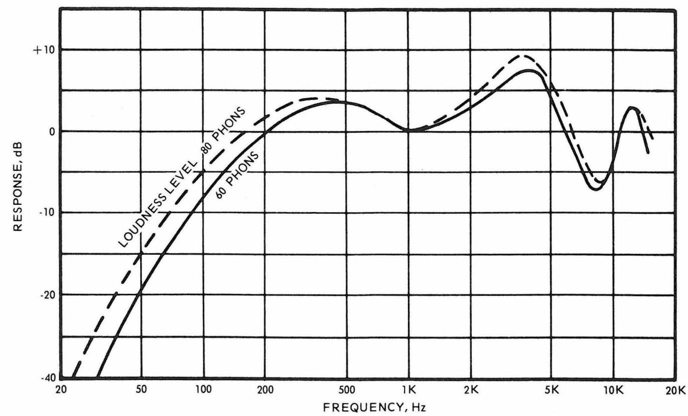

Fig. 4-7. The frequency response of human ear is not flat. Further, the ear's response varies with loudness of the sound, being flatter for very loud sounds than for soft sounds. The loudness control on hi-fi amplifiers is intended to compensate somewhat for this characteristic of the ear.

LOUDNESS CONTROL

Let us assume that the high fidelity enthusiast adjusts the volume control on his amplifier so that the level of the recorded symphony music is pleasing as a background to conversation (assumed to be about 60 phons). As the passage was played at something like an 80 phon loudness level in the concert hall, something needs to be done to give the bass and treble of the music the proper balance at the lower-than-concert-hall level. Our enthusiast would find it necessary to increase both bass and treble for good balance. The loudness control found on many amplifiers adjusts electrical networks to compensate for the change in frequency response of the ear for different loudness levels. But the curve corresponding to a given setting of the loudness control applies only to a specific loudness level of reproduced sound. To be of much help, loudness controls must be calibrated for each set of equipment, each room, etc. This discriminating audiophile, however, may not be very happy if this loudness control compensation is locked to the position of the volume control which is not necessarily closely related to the perceived loudness of the free floating sound. How can one simple loudness control care for all these variables? So, perhaps we have not reached the ultimate in the design of a loudness control. The volume setting is affected by the output of the phono cartridge, the

sensitivity of the power amplifiers, the efficiency of the loudspeakers, as well as the vagaries of human hearing and room acoustics.

LOUDNESS METER

Much effort in the broadcasting industry has gone into development of a loudness meter which would give an indication proportional to the subjective response of people. Are radio and television commercials actually louder than the program material? The sponsors say "no", the consumer says "yes". The loudness meter is supposed to give an unequivocable meter reading to settle the question. Specifically, such a loudness meter must divide the signal spectrally into multiple channels (as the ear does), rectify the signals of each channel, perform a loudness summation and loudness integration and provide a suitable readout [6]. Although tests indicate that this loudness meter does truly measure loudness with reasonable accuracy, the commercial loudness problem seems to be with us still.

LOUDNESS AND BANDWIDTH

As far as human reaction is concerned, a wideband noise is louder than a tone of the same sound pressure level. If a filter is introduced so that the high and low cutoff points are adjustable we find that narrowing the bandwidth reduces the loudness of the noise to a certain point, but further reduction in bandwidth beyond this point has no effect on loudness. [7] This bandwidth, below which no reduction in loudness is found, is called a critical band. If a bandwidth of noise centered on 1 kHz is progressively narrowed, the width of the critical band is found to be about 160 Hz. At 5,000 Hz the critical bandwidth determined in this way is about 900 Hz. Below 500 Hz the critical bandwidth flattens off at about 100 Hz. A plot of the critical bandwidths as a function of frequency is shown in Fig. 4-8.

The concept of critical bands of the ear is established in other ways. For example, in masking experiments in which tones are masked by wideband noise, it is found only that noise within the critical band centered on the frequency of the tone is effective in masking the tone. The tone is masked when the power in the noise band equals that of the tone. Other experiments involving thresholds and phase relations further establish the critical band concept. It appears that an explanation of critical band action is to be found in the inner ear, in the mechanics of the basilar membrane.

The ear-brain system is a spectrum analyzer in which the critical bands have a part. Their resolution, however, does not appear to be sufficient to account for the very fine pitch discrimination of which the ear is capable.

In Fig. 4-8 the bandwidth of ⅓ octave filters is plotted for comparison with critical bandwidths. The agreement is not perfect, but sufficient to support the use of ⅓ octave filters in many acoustical measurements bearing on human hearing in one form or another.

LEVEL DISCRIMINATION

Modern faders are of the composition type giving infinitely small graduations in level as they are adjusted. However, the wirewound faders on old fashioned mixing consoles were commonly built with 2 dB steps. If these steps were 5 dB, movement of the fader would give very noticeable step-wise increments in sound that would be very disturbing. If they were 0.5 dB steps, the fader would be too expensive. The 2 dB steps were selected because steps of this magnitude are generally just barely detectable by an expert ear. This is only approximately true. Detecting differences in intensities varies somewhat with frequency and also with sound level.

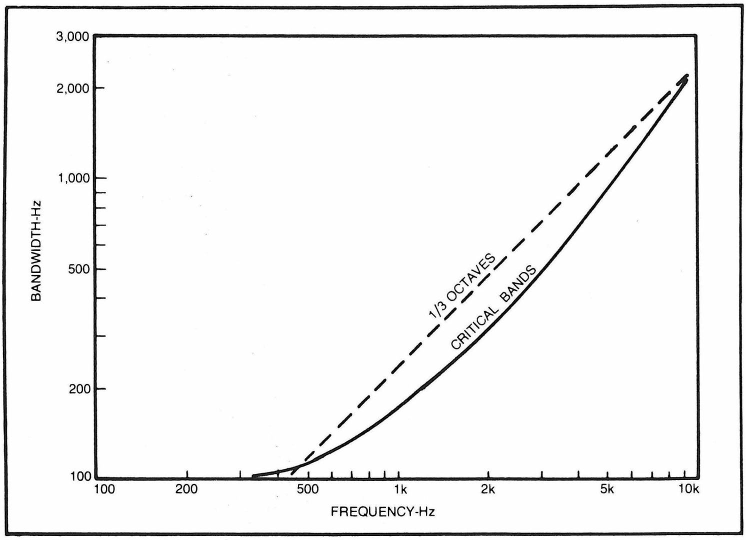

Fig. 4-8. The width of critical bands of the human ear is here compared to ⅓ octave bandwidths. Although the agreement is not perfect, it is sufficient to support the use of ⅓ octave filters in many acoustical measurements related to hearing.

At 1 kHz, for very low levels, a 3 dB change is the least detectable by the ear but at high levels the ear can detect a 0.25 dB change. For a very low level 35 Hz tone, 9 dB is the least detectable. However, for the important midfrequency range and for commonly used levels, the minimum detectable change in level that the ear can detect is about 2 or 3 dB. Making level adjustments less than this is wholly unnecessary.

PITCH DISCRIMINATION

Those who work in sound are interested in the ear's power of discrimination in pitch as well as intensity. Experimenters have found that with moderately loud sounds below 1 kHz we can detect a change in frequency of about 3 Hz. Above 1 kHz the minimum detectable change in frequency turns out to be a constant percentage of the frequency and amounts to about one semitone of the musical scale.

Researchers tell us that there are about 280 discernible steps in intensity and some 1,400 discernible steps in pitch that can be detected by the human ear. As changes in intensity and pitch are the very stuff of communication, it would be interesting to know how many combinations are possible. Offhand, it might seem that there would be 280 × 1,400 = 392,000 combinations detectable by the ear. This is overly optimistic because the tests were conducted by comparing two simple, single-frequency sounds in rapid succession and bears little resemblance to the complexities of commonly heard sounds. More realistic experiments show that the ear can detect only about 7 degrees of loudness and 7 degrees of pitch or only 49 pitch-loudness combinations. This is not too far from the number of phonemes (the smallest unit in a language that distinguishes one utterance from another) which can be detected in a language.

THE EAR AS AN ANALYZER

Let us perform an experiment. Listening to a good symphony orchestra in your favorite concert hall, concentrate your attention on the first violins. Now focus attention on the clarinets, then the percussion section. Next listen to a male quartet and single out the first tenor, the baritone, the bass. This is a very remarkable power of the human ear/brain combination. In the ear canal all these sounds are mixed together; how does the ear succeed in separating them? The sea surface may be disturbed by many wave systems, one due to local wind, one from a distant storm, and several wakes from passing vessels. The eye cannot separate these, but this is

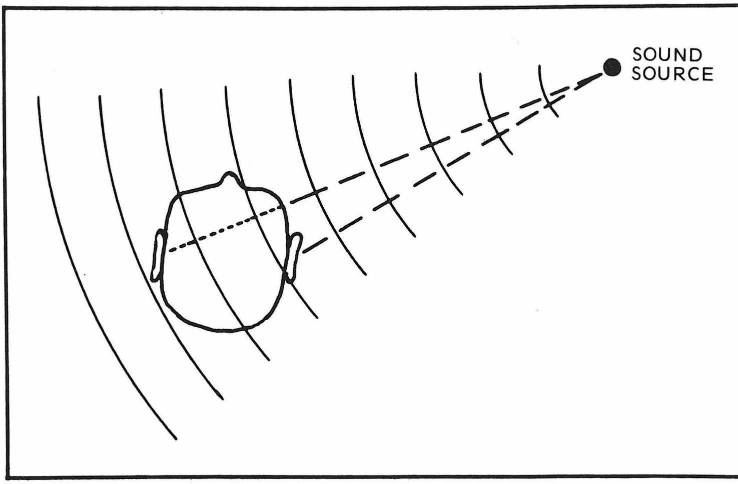

Fig. 4-9. Our binaural directional sense is dependent in part on the difference in intensity and phase of the sound falling on the two ears.

essentially what the ear is constantly doing with complex sound waves. In fact, by rigorous training a keen observer can listen to the sound of a violin and pick out the various overtones apart from the fundamental!

PERCEPTION OF DIRECTION

Stereophonic records and sound systems are a relatively new development. Stereo hearing has been around at least as long as man. Both are concerned with the localization of the source of sound. In early times some people thought that having two ears was like having two lungs or two kidneys, if something went wrong with one the other could still function. Lord Rayleigh laid that idea to rest by a simple experiment on the lawn of Cambridge University. A circle of assistants spoke or struck tuning forks and Lord Rayleigh in the center with his eyes closed pointed to the source of sound with great accuracy, confirming the fact that two ears function together in binaural localization.

It turns out that actually two factors are involved, the difference in intensity and the difference in time of arrival (phase) of the sound falling on the two ears. In Fig. 4-9 the ear nearest the source receives a greater intensity than the far ear because the hard skull increases sound pressure on the near side and the "sound shadow" (diffraction) caused by the head reduces sound pressure on the far side. Because of the difference of distance to the source, the far ear

receives sound somewhat later than the near ear. Below 1 kHz the phase effect dominates while above 1 kHz the intensity effect dominates. There is one localization blind spot. A listener cannot tell whether sounds are coming from directly in front or from directly behind because the intensity of sound arriving at each ear is the same and in the same phase.

Another method of perception of direction comes into play in a relatively small room. The sound reaches the person over a direct path followed by many reflections from many different directions. The sound that arrives first creates in the hearer the main perception of direction. This has been called "the law of the first wavefront".

BINAURAL EFFECTS

In World War I aircraft detection and location systems utilized the binaural effect. A pair of horns picked up the sound which was conducted to the ears by stethoscope tubes, left horn to left ear, right horn to right ear. By exaggerating the separation of the horns, the phase effect could be heightened and sharper bearings could be obtained. The American "S-Boat" submarines in World War I had a similar system for underwater detection. In place of horns, rubber spheres, separated several feet and in contact with the water connected to stethoscope tubes within the hull.

It is interesting that two ears hear more than one. A sound striking two ears seems louder than a sound in one ear (though not twice as loud). Although opening and closing one eye does not change the apparent brightness of the scene, the outputs of the two ears do add.

When we are in a noisy room or one plagued with too much reverberation, we are less irritated by the defects than when we hear sounds over a "one-eared" (monaural) channel such as radio or television. In the same room our binaural sense of direction can be brought into play, concentrating on the speaker and rejecting noise, echoes, and reverberation. This is one good reason radio and television studios as well as recording studios handling monaural program material must have good acoustics and low noise levels.

PERCEPTION OF SINGLE REFLECTIONS (HAAS EFFECT)

Haas[8], in 1951, found that a simulated reflection was not heard as a separate sound unless the delay exceeded about 50 milliseconds. Haas used continuous speech on two loudspeakers, the input signal to one could be attenuated or delayed at will. He asked

the subjects to adjust the attenuator to one loudspeaker until the undelayed and delayed signals were of equal loudness. Haas was amazed at what he found. For delays in the region of 5 to 30 ms the delayed loudspeaker level had to be 10 dB greater to sound as loud as the undelayed. Only for delays greater than 40 or 50 ms did the delayed signal take on the character of a discrete echo. For the shorter delays an apparent fusion effect dominated as illustrated in Fig. 4-10. The delayed signal (reflection) was combined with the undelayed, increasing the apparent level of the combination. Along with the increase in loudness was also a pleasant change in sound quality. We see, then, that "echoes" returned within the 25-30 ms limit are not perceived as discrete echoes at all but contribute to loudness and improvement of sound quality. For delays greater than 50 ms the delayed signal is perceived as a discrete echo and is very disturbing.

This fusion effect, first noted by Joseph Henry in 1854 and studied by many during the intervening years, is confirmed daily in the workaday world of audio. The broken line of Fig. 4-10 shows roughly the confirming results of Meyer and Schodder as described by Kuttruff. [9]

PERCEPTION OF MULTIPLE REFLECTIONS (SERAPHIM EFFECT)

The fusion zone of Fig. 4-10 can be exploited in audio work in

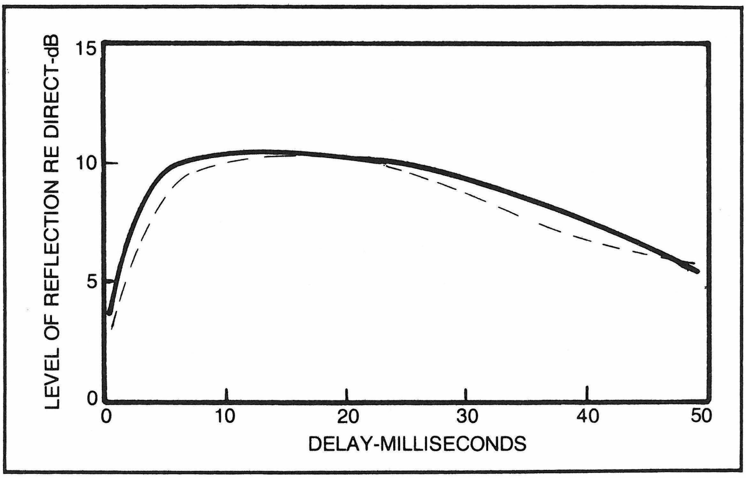

Fig. 4-10. The definitive curve of the "Haas Effect" in which reflections delayed 5-25 ms increase loudness and improve sound quality. For delays less than 25 ms the reflections are not perceived as echoes, for delays greater than 50 ms they are heard as discrete echoes.

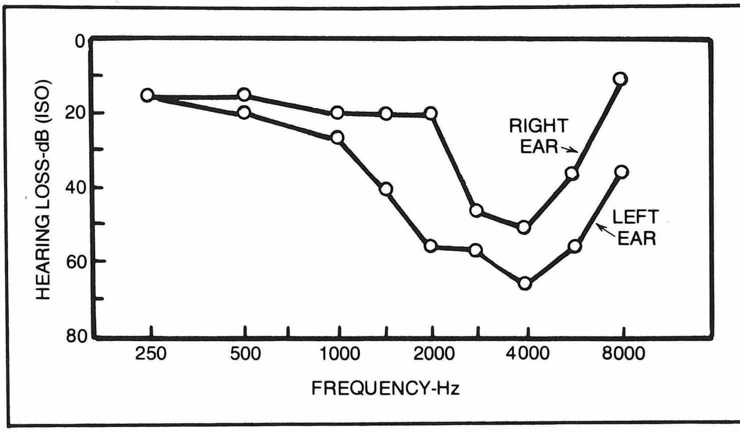

Fig. 4-11. Audiograms showing serious loss centered on 4 kHz, presumably resulting from years of exposure to high level sound in a recording studio control room.

ways ranging from sound reinforcement systems to control room design. There are times when one might wish the fusion zone were wider. The work of Seraphim (conveniently reported by Kuttruff because the original paper is in German) [9] has shown that with a carefully spaced succession of reflections the fusion zone can be extended to 50 ms (2 reflections), 70 ms (3 reflections), or 80 ms (4 reflections). These delayed signals (reflections) were all from the front. Real life reflections may come from the side or from behind in which case the effect tends to disintegrate.

HEARING—A PRECIOUS COMMODITY

There seems to be an increasing awareness that loud sounds can damage hearing. This may more often be an intellectual awareness accompanying an "it can't happen to me" attitude. Permanent hearing losses are being added to slowly and insiduously in the control rooms of recording studios, at rock concerts, discos, and (in isolated areas or with forgiving neighbors) in the hi-fi listening room. No longer is hearing loss associated only with workers with heavy machinery and in boiler factories.

Even when organ transplants are as common as face lifts, some very special problems will be associated with transplanting the chick-pea sized cochlea imbedded in solid skull bone and in splicing the aural nerve. For the time being we must operate on the principal that these are the only ears we will ever have and when hair cells are destroyed, something is lost forever.

The key to conservation of hearing is the audiogram. Comparing todays audiogram with one taken a couple of years ago establishes the trend; if downward, what can be done to check it? The audiogram of Fig. 4-11, which looks something like the Big Dipper constellation is that of a 50-ish sound mixer in a recording studio. The indications are that this loss, centered on 4 kHz, is the accumulation of many years of listening to high level sounds. Would you like to have audiograms like these?

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5

Resonances in Rooms

Striking a bell causes it to vibrate at its "natural" frequency. A well-placed kick can set a suspended steel plate vibrating at a frequency determined by its physical properties. In fact, a 3 × 6-foot steel plate has been used in a very successful artificial reverberation device but in this instance care was taken to operate at frequencies far removed from the natural frequencies of the plate. The hardwood bars of a xylophone vibrate at their natural frequencies determined by their size. Seismic disturbances in the earth occur at frequencies far below the range of the human ear because of the great size of the earth. At the California Institute of Technology the author heard a tape recording of such earth vibrations played back at a speed far higher than the recording speed. The earthquake had set the earth ringing like a bell!

Children are delighted to hear "the roar of the ocean" by holding a large conch shell to their ears. And who hasn't made a bottle "sound its A" by blowing across its mouth much as the jug player in a hillbilly band? Alas! Probably they do not realize the great acoustical significance of their actions! The spiral chamber of the great sea shell and the cavity within a jug are resonators springing to life in response to a bit of sound energy at the frequency at which the enclosed air is resonant. In the case of the conch shell, those components of the prevailing noise near this natural frequency are selectively reinforced, producing the "ocean's roar." The stream of air across the mouth of the jug provides the energy

which sets the air within singing the characteristic resonant tone of the jug.

RESONATORS

Hermann von Helmholtz (1821-1894) performed some interesting acoustical experiments with resonators functioning much as the conch shell and the jug. His resonators were a series of metal spheres of graded sizes, each fitted with a neck, appearing somewhat like the round-bottom flask found in the chemistry laboratory. In addition to the neck there was another small opening to which he applied his ear. The resonators of different sizes resonated at different frequencies and by pointing the neck toward the sound under investigation he could estimate the energy at each frequency by the loudness of the sound of the different resonators.

There were numerous applications of this principle long before the time of Helmholtz. There is evidence that bronze jars were used by the Greeks in their open-air theaters to provide some artificial reverberation. A thousand years ago Helmholtz-type resonators were imbedded in church walls in Sweden with the mouths flush with the wall surface, apparently for sound absorption[1]. The walls of the modern sanctuary of Tapiola Church[2] in Helsinki, Finland, are dotted with slits in the concrete blocks (Fig. 5-1). These are

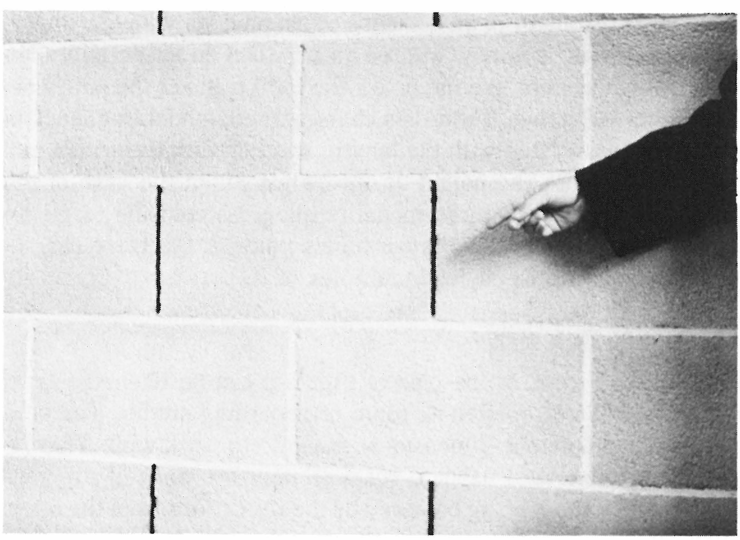

Fig. 5-1. Helmholtz-type resonators built into the wall of the Tapiola Church in Helsinki, Finland. The slots and cavities behind them act both as absorbers and diffusers of sound.

resonator "necks" which open into cavities behind, together forming resonating structures. Energy absorbed from sound in the room causes each resonator to vibrate at its own characteristic frequency. Part of the energy is absorbed, part reradiated. The energy reradiated is sent in every direction, contributing to the diffusion of sound in the room. The resonator principle, old as it is, continually appears in modern, up-to-the-minute applications and we shall soon be applying it in the acoustical design of listening rooms and studios.

BATHROOM ACOUSTICS

Why is it that singing in the shower or tub is such a satisfying experience (to the singer, at least)? Because here one's voice sounds richer, fuller, and more powerful than anywhere else! The case of the bathroom baritone clearly illustrates the effect of resonance in a small room and the resulting reinforcement of sound at certain frequencies related to the dimensions of the room. Exciting the air in the bathroom at frequencies far removed from these characteristic frequencies results in weaker sounds, except at multiples of these frequencies, where the effect may be very much like that at the lowest natural frequencies.

The man singing in the bathroom is, in a sense, inside a Helmholtz resonator or an immense organ pipe but with one important difference; it is now a three rather than an essentially one-dimensional system like the pipe. The hard walls of the bathroom are highly reflective. There is a characteristic modal frequency of resonance associated with the length, another with the width, and still another with the height of the bathroom. In the case of the cubical bathroom, all three modal frequencies coincide to give a mighty reinforcement to the baritone's voice at the basic characteristic modal frequency and multiples of it.

RESONANCE IN A PIPE

The two ends of the pipe of Fig. 5-2 can be likened to two opposing walls of a listening room or recording studio. The pipe gives us a simple one-dimensional example to work with. That is, we can examine what happens between opposite walls of a rectangular room without being bothered by the reflections from the other four surfaces. The pipe, closed at both ends and filled with air, is a resonator capable of vibrating at its characteristic frequencies when excited in some way. Air inside the organ pipe may be set to

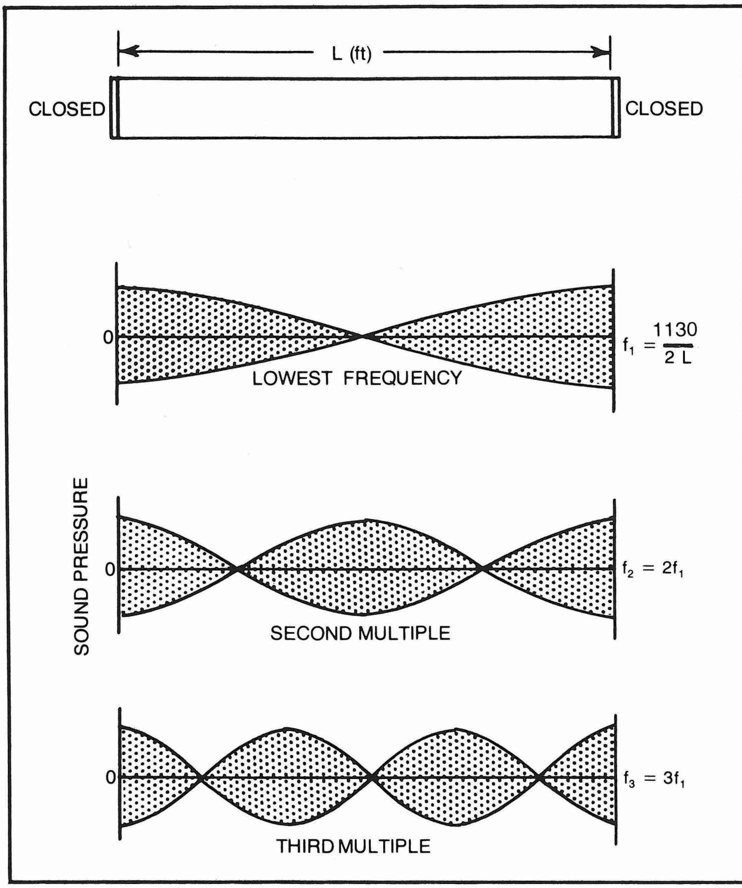

Fig. 5-2. A pipe closed at both ends helps us to understand how resonance occurs between two opposing walls of a listening room or studio. The distance between the walls determines the characteristic frequency of resonance.

vibrating by blowing a stream of air across a lip at the edge of the pipe. It is simpler for us to place a small loudspeaker inside the pipe. Let us feed a sine wave signal to the loudspeaker and vary the frequency. If we drill a small hole in the pipe in the end opposite the loudspeaker and place our ear against it we can hear the low-level tones radiated by the loudspeaker. As the frequency is increased, nothing much is noted until the frequency radiated from the loudspeaker coincides with the natural frequency of the pipe. At this frequency, f_1, modest energy from the loudspeaker is strongly reinforced and a relatively loud sound is heard at the earhole. As the frequency is increased, the loudness is again low until a frequency of

$2f_1$ is reached, at which point another strong reinforcement is noted. Such resonant peaks can also be detected at $3f_1$, $4f_1$ . . . etc.

Now let us assume that we have the means of measuring and recording the sound pressure all along the pipe. In Fig. 5-2 the graphs below the sketch of the pipe show how the sound pressure varies along the length of the pipe under different conditions of excitation. A sound wave traveling to the right is reflected from the right plug and a sound wave traveling to the left from the left plug. The left-going waves react with the right-going waves to create, by superposition, a **standing** wave at the natural frequency of the pipe or one of its multiples. Measuring probes inserted through tiny holes along the pipe could actually measure the high pressure near the closed ends and zero at the center, etc. Similar nodes (zero points) and antinodes (maxima) can be observed at $2f_1$, $3f_1$, $4f_1$ . . . etc., as shown in Fig. 5-2. The dimensions of a studio or listening room determine the characteristic frequencies of a rectangular room much as though we had a north-south pipe, an east-west pipe, and a vertical pipe, the pipes corresponding to the length, width, and height of the room, respectively.

AN EXPERIMENT

Here is an experiment the reader can easily perform to illustrate the existence and nature of standing waves in a room. Connect a signal generator to the input of an amplifier driving a loudspeaker located in the room. Adjust the signal generator to 1,000 Hz and set the gain for a reasonably comfortable level of sound. We have seen (Eq. 1-3) that the wavelength of sound in air can be calculated by dividing the speed of sound by the frequency of the sound under consideration, in this case $1130/1000 = 1.13$ ft. From this we know that the room is many wavelengths long, wide, and high.

A very complex sound field is observed. Moving the head causes wide fluctuations in sound loudness. For this experiment, closing one ear makes good sense because our desire is to explore the sound field changes throughout the room rather than what happens in our head as two ear signals are combined. These variations of loudness of the 1,000 Hz tone observed by moving about the room are the result of the superposition of the sound directly from the loudspeaker with reflections from all six surfaces of the room. The location of the peaks and dips do not change as long as the frequency of sound is held constant and sound energy continues to flow out from the loudspeaker. These are standing waves: sta-

tionary effects resulting from the dynamic interaction of various components of sound in the room.

WAVES OR RAYS?

The classical work on reverberation of sound in auditoriums was based purely on statistical and geometrical concepts. Sound energy was assumed to be distributed uniformly throughout the room. Predictions of performance in large auditoriums based on the ray approach agreed fairly well with measurements. However, it is found that all is not well when the ray approach is applied to smaller rooms. Calculations based on ray theory and actual measurements are often far apart. One of the reasons for this is that the sound energy is not uniformly distributed. In the case of the pipe with closed ends we see that the sound energy may be anything but uniformly distributed. For smaller listening rooms and recording studios, resonance effects may grossly affect the distribution of sound energy. Thus in the case of small enclosures we are faced with the necessity of considering both waves (resonance and standing wave effects) and rays to get a complete picture of the acoustical sitution.[3, 4]

RESONANCES IN LISTENING ROOMS AND SMALL STUDIOS

The fact that the human ear spans a tremendous ten octaves aggravates the acoustical problems in small enclosures. Let us consider a sound at each end of the 10-octave frequency range of the ear, one at 20 Hz and another at 20 kHz. At the low end a studio 28 feet long is resonant in this lengthwise mode at a frequency of 1130/(2)(28) or 20 Hz. At this frequency and for a room of this size we have no alternative but to approach the acoustical analysis of the room from the standpoint of standing waves and resonances.

At 20 kHz, however, the wavelength is only about ⅝ inch, which is very small compared to the 28-foot length of the room. While, theoretically, a standing wave having a wavelength of ⅝ inch could be maintained between the ends of the 28-foot room, practically speaking, small irregularities of the surfaces would tend to diffuse the sound and thus destroy the resonance effect. Further, in trying to detect the standing wave there would be many problems because microphone diaphragms, eardrums, etc., are large compared to the wavelength and would thus tend to average rather than delineate resonant reinforcements and cancellations.

The transition from wave acoustics at the low end of the audible spectrum to ray acoustics at the high end is a very indefinite one. Very roughly we can say that for frequencies less than about 300 Hz, ray acoustics are of very limited utility. When the low frequency resonances in small rooms are adequately controlled in the general interests of sound quality, diffusion of sound is usually sufficient to give reasonable validity to reverberation calculations based on the ray approach.

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6

Standing Waves in Listening Rooms and Small Studios

"Colorations" due to room resonances* (normal modes) are often very serious with voice. Such colorations take the form of unnatural and monotonous emphasis of certain frequencies in the speaker's voice, giving it an unpleasant "roughness." Room colorations can also affect music but are most difficult to isolate because of the transient and non repetitive nature of most music. Also, the average listener knows better how a voice should sound than a given musical selection. Standing wave effects can accentuate particular music notes or voice components as much as 8 or 10 dB.

In a listening room or studio a mode is heard as a coloration of the desired sound through a tendency toward reinforcement at the modal frequency. If a certain modal frequency is isolated from its neighbors, its effect is more likely to be audible. At the higher frequencies, say above about 300 Hz, individual modes are seldom distinguishable, but below this frequency, isolated modes are often troublesome in small rooms like studios and living rooms.

There is a tendency for acoustical faults of a room to be rejected by the binaural hearing mechanism of a person in that room (or hearing a stereo rendition of the sound which originated in that room). Because the degree of this binaural compensation is limited, it must not be used as an excuse for not correcting listening room and studio acoustics.

*Perhaps a more accurate term would be "characteristic frequency". However, because these normal modes exhibit resonance-like "tuning curves" with bandwidth dependent on losses, the concept of room resonances will be used throughout this book.

The volume of air enclosed in a room is a complex vibratory system. To understand the acoustics of small listening rooms and studios is to understand this vibratory system and the large number of characteristic frequencies (modal frequencies) associated with it.

WAVE ACOUSTICS

The desire for simplicity in this book must not overshadow our desire for accuracy and completeness of understanding. A physicist, in approaching the problem of sound distribution in a room such as a studio or listening room, would instinctively turn to the partial differential mathematical statement dubbed "the wave equation", and for very good reasons. While we may circumvent the mathematical steps, there is no getting around the specific solution of the wave equation applying to rectangular enclosures.

The length (L), width (W), and height (H) of our rectangular room are related to the mathematician's x, y, and z axes as shown in the sketch of Fig. 6-1. The longest dimension (L) is placed along the x-axis, the next longest (W) along the y-axis, and the shortest (H) along the z-axis. This is simply for convenience in keeping track of things. The solution of the partial differential wave equation for this rectangular room gives the frequencies of the normal modes of the room:

$$\text{Frequency} = \frac{c}{2}\sqrt{\frac{p^2}{L^2} + \frac{q^2}{W^2} + \frac{r^2}{H^2}} \qquad \textbf{(6-1)}$$

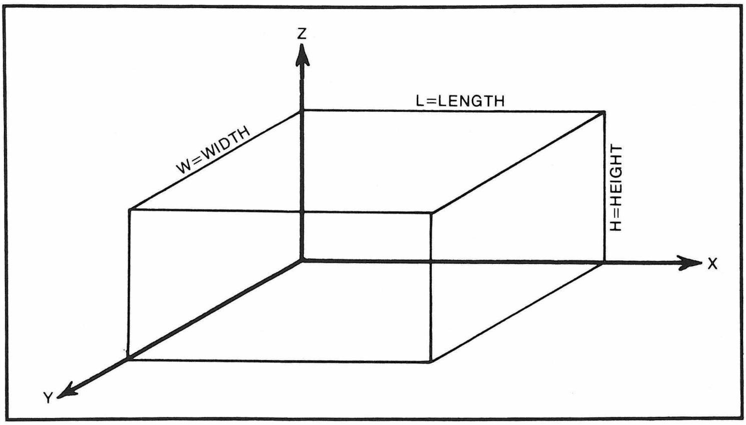

Fig. 6-1. Orientation of the room to be analyzed with respect to the x, y, and z coordinates of the mathematician.

where:

c	=	speed of sound, about 1130 ft/sec
L	=	length of room, ft
W	=	width of room, ft
H	=	height of room, ft
p, q, r	=	integers 0, 1, 2, 3, 4 . . . etc . .

Rather than skipping this section as being "too mathematical", hold on for a while longer to discover just how revealing Eq. 6-1 really is and how helpful it can be in understanding the strange acoustical antics of a roomful of air.

The organ pipe analogy of Fig. 5-2 is covered by Equation 6-1. If we imagine, for the moment, that the east and west walls and the ceiling and floor of our room are eliminated, only the opposite and parallel north and south walls remain to affect the sound we introduce between them. At a certain frequency, f_1, a standing wave will be set up between them such as shown in Fig. 5-2. This occurs along one of the three axes of the room, hence it is called an axial mode. With the assumption that the length (L) of the room is the distance between the north and south walls, we can see just how Equation 6-1 fits this situation. It is obvious that any of the three terms under the square root sign may be eliminated by assigning a zero for p, q, or r. In our case we do this for the width (q = 0) and the height (r = 0) to isolate the length (L) for consideration. For L we assign the integer p = 1. This reduces Equation 1 to:

$$\text{Frequency} = \frac{c}{2}\sqrt{\frac{1^2}{L^2}}$$

which further simplifies to:

$$\text{Frequency} = \frac{c}{2L} \qquad \textbf{(6-2)}$$

Substituting 1130 ft/sec for c:

$$\text{Frequency} = \frac{1130}{2L} = \frac{565}{L} \qquad \textbf{(6-3)}$$

This says that the f_1 frequency (p=1) for the axial mode corresponding to the length L is 565 divided by the length of the room in feet. Another axial mode for length L exists for p = 2:

$$\text{Frequency} = \frac{c}{2}\sqrt{\frac{2^2}{L^2}} = \frac{2c}{2L} = 2\left(\frac{565}{L}\right)$$

And for p = 3:

$$\text{Frequency} = 3\left(\frac{565}{L}\right)$$

and we note the similarity to the f_1, $2f_1$, $3f_1$, etc., for the organ pipe of Fig. 5-2.

We conclude that putting p equal to 1 determines the lowest frequency of resonance supported by the two plane, parallel reflecting surfaces, the north and south walls of our room. Integers of $p = 2$ and $p = 3$ describe the second and third multiple of this axial mode, and so on.

By assigning $p = 0$ to the length (L) and $r = 0$ to the height (H), and $p = 1$ to the width (W), the lowest frequency of the width axial mode can be calculated. In a similar way the lowest modal frequency associated with the height (H) of the room can be found. By assigning the higher integers 2, 3, 4, etc., to p the successively higher multiple frequencies associated with any axial mode can be found. It is not right to call the multiples of f_1 harmonics of f_1. The term harmonic is reserved for the components of a distorted, non-sinusoidal, periodic wave. We note with interest that sound energy in the room "in the cracks" between these natural frequencies associated with length, width, or height of the room will not, as far as axial modes are concerned, find a resonant boost.

THREE KINDS OF MODES

Ah! But now we shall see why it was necessary to wade through the intricacies of Equation 6-1. What happens if $p = 1$, $q = 1$, and $r = 0$? Only the length (L) and the width (W) terms of Eq. 6-1 need be considered, the height (H) term drops out. When only one integer is zero we do not have an axial mode any more, but a mode involving two pairs of surfaces. This is called a *tangential mode*. Similarly, if there are no zeros at all, all three pairs of plane parallel walls are involved and what is called an *oblique mode* results. There is a modal frequency associated with each combination of p, q, and r as shown in Table 6-1.

Don't let the digital appearance of Table 6-1 fool you! The lowest frequencies are associated with p, q and/or r being zero or 1. But we have learned from the organ pipe (Fig. 5-2) that multiples of these frequencies are also supported by the rectangular room. From

Mode	p	q	r
Axial (N-S)	1	0	0
Axial (E-W)	0	1	0
Axial (vertical)	0	0	1
Tangential	1	1	0
Tangential	1	0	1
Tangential	0	1	1
Oblique	1	1	1

Table 6-1. Identifying Modes.

here on things do not particularly get more complicated, but they do get more complex as shown in Table 6-2.

So it is important for us to mind our p's and q's (and r's)! All modes are identified by the p's, q's, and r's (i.e., whether axial, tangential, or oblique) and by inserting the appropriate integer values and room dimensions in Eq. 6-1, the frequency of any mode can be found.

Before exploring modal frequencies in detail, it is well to get some sort of physical picture of tangential and oblique modes. Figure 6-2 shows that the axial modes involve only two surfaces of the room, tangential modes four surfaces, and oblique modes all six surfaces. In the highly simplified sketch of Fig. 6-2 the arrows appear to be rays of sound. Actually, we are into waves in Eq. 6-1. It would be closer to consider the arrows as directions the wave fronts are following rather than rays.

MODAL FREQUENCIES OF A TYPICAL ROOM

There is no better way to get a firm grip on this very important modal concept than to take a typical rectangular room and compute axial, tangential, and oblique modal frequencies. The room selected is what one might call a "carefully selected typical example". The length, width, and height are proportioned according to techniques to be considered in Chapter 7. The dimensions selected are: 23.3 × 16 × 10 ft. This gives a volume of 3,728 cu. ft., well over the recommended practical minimum of 1,500 cu. ft. This room may be

Table 6-2. Further Examples of Model Configurations.

Mode	p	q	r
None (no walls!)	0	0	0
Axial (E-W)	0	1	0
Tangential	1	2	0
Tangential	3	0	1
Oblique	1	2	3
Oblique	4	1	1

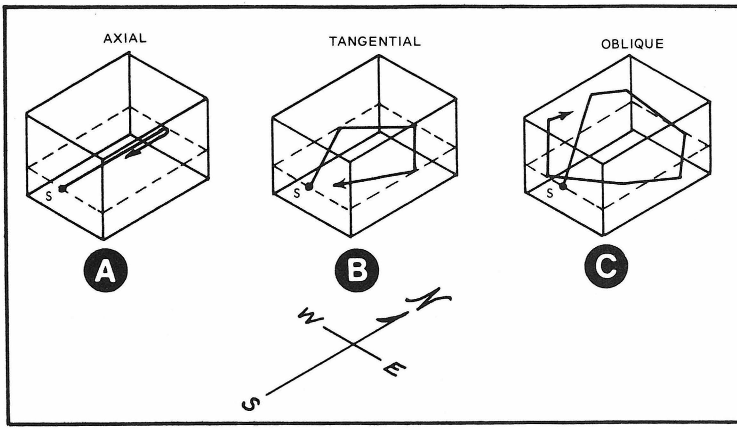

Fig. 6-2. There are three basic standing wave patterns that can form in a room. The axial mode (A) involves reflections from only one pair of surfaces. The tangential mode (B) involves two pairs of surfaces. The oblique mode (C) involves all six surfaces. The axial modes are of the greatest practical significance in small listening rooms and studios. The arrows should be considered the directions the wave fronts are traveling and not rays of sound.

thought of as a small studio, living room hi-fi center, a control room, or a hi-fi equipment dealer's demonstration room. The first step in treating such rooms acoustically is proportioning dimensions to distribute the modal frequencies, after which comes the treatment of surfaces to achieve the proper reverberatory characteristics. The example chosen is a very practical room in which we are vitally interested in the distribution of modal frequencies which control the acoustical quality of the space.

We have L = 23.3 ft, W = 16 ft, and H = 10 ft to substitute in Eq. 6-1. By assigning various integers to p, q, and r we can calculate axial, tangential, and oblique modal frequencies which can then be scrutinized carefully. Figure 6-3 illustrates the commonly accepted method of describing a mode. For example, consider the 0, 1, 0 mode

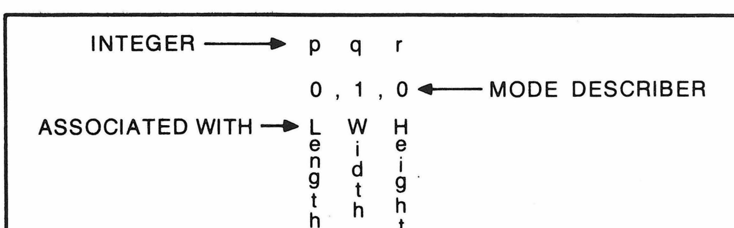

Fig. 6-3. The accepted form for describing a normal mode by the integers p, q, and r in Equation 6-1.

L W H	Ax	Tn	Ob
1 0 0	24.18		
0 1 0	35.27		
1 1 0		42.76	
2 0 0	48.37		
0 0 1	56.43		
2 1 0		59.86	
1 0 1		61.39	
0 1 1		66.55	
0 2 0	70.54		
1 1 1			70.80
3 0 0	72.55		
2 0 1		74.32	
1 2 0		74.57	
3 1 0		80.67	
2 1 1			82.27
2 2 0		85.53	
0 2 1		90.33	
3 0 1		91.92	
1 2 1			93.51
4 0 0	96.74		
3 1 1			98.45
3 2 0		101.19	
2 2 1			102.47
4 1 0		102.97	
0 3 0	105.81		
1 3 0		108.54	
4 0 1		111.99	
0 0 2	112.86		
1 0 2		115.42	
3 2 1			115.86
2 3 0		116.34	
4 1 1			117.42
0 1 2		118.24	
4 2 0		119.72	
0 3 1		119.91	
1 1 2			120.69
1 3 1			122.33
2 0 2		122.79	
2 1 2			127.75
3 3 0		128.29	
2 3 1			129.30
4 2 1			132.36

L W H	Ax	Tn	Ob
0 2 2		133.09	
3 0 2		134.17	
1 2 2			135.27
3 1 2			138.73
3 3 1			140.16
0 4 0	141.08		
2 2 2			141.61
1 4 0		143.13	
4 3 0		143.36	
4 0 2		148.65	
2 4 0		149.14	
3 2 2			151.58
0 4 1		151.94	
4 1 2			152.77
1 4 1			153.86
4 3 1			154.07
0 3 2		154.70	
1 3 2			156.58
3 4 0		158.64	
2 4 1			159.46
2 3 2			162.09
4 2 2			164.53
3 4 1			168.38
0 0 3	169.29		
3 3 2			170.87
1 0 3		171.01	
4 4 0		171.06	
0 1 3		172.93	
1 1 3			174.61
2 0 3		176.07	
2 1 3			179.56
4 4 1			180.13
0 4 2		180.67	
1 4 2			182.28
4 3 2			182.46
0 2 3		183.40	
3 0 3		184.18	
1 2 3			184.99
2 4 2			187.03
3 1 3			187.53
2 2 3			189.67
3 4 2			194.69

L W H	Ax	Tn	Ob
4 0 3		194.98	
3 2 3			197.23
4 1 3			198.15
0 3 3		199.64	
1 3 3			201.10
4 4 2			204.93
2 3 3			205.41
4 2 3			207.35
3 3 3			212.41
0 4 3		220.37	
1 4 3			221.69
4 3 3			221.84
2 4 3			225.61
0 0 4	225.72		
1 0 4		227.01	
0 1 4		228.46	
1 1 4			229.74
2 0 4		230.85	
3 4 3			232.00
2 1 4			233.52
0 2 4		236.49	
3 0 4		237.10	
1 2 4			237.72
3 1 4			239.70
4 4 3			240.67
2 2 4			241.38
4 0 4		245.58	
3 2 4			247.37
4 1 4			248.10
0 3 4		249.29	
1 3 4			250.46
2 3 4			253.94
4 2 4			255.51
3 3 4			259.63
0 4 4		266.18	
1 4 4			267.28
4 3 4			267.40
2 4 4			270.54
3 4 4			275.89
4 4 4			283.22

Data prepared by Charles Nairn

Table 6-3. Modal Frequencies for Favorably Proportioned Room. 23.3 x 16 x 10 Ft. 1,0,0, to 4,4,4

in this brand of shorthand. The first zero means p = 0, the next 1 means q = 1, and the final zero means r = 0. For a physical picture of this mode, the lengthwise (N – S) and vertical terms drop out because of the zeros and the axial mode corresponding to the room width is isolated for study. Another example: the 1, 2, 3 mode is an oblique mode with p, q, and r integers being 1, 2, and 3, respectively.

In selecting p's, q's, and r's for our calculations, some sort of lid needs to be placed on the number of integers lest excessive computation time is required. To place four integers (0,1,2,3) in all possible combinations of p's, q's, and r's gives 4^3 or 64 separate solutions of Eq. 6-1. For five integers (0,1,2,3,4) there are $5^3 = 125$ solutions and for six integers, 6^3 or 216 solutions. It is obvious that solving Eq. 6-1 by hand becomes a bit of a chore and it is really a job for a computer.

Through the good graces of Charles Nairn of Communications Technology, Inc., of Detroit, Michigan, Eq. 6-1 was solved in 12 minutes for all 125 combinations for integers 0,1,2,3 and 4 substituted for p, q, and r. Table 6-3 was photographed directly from his printout which assembled the modal frequencies in ascending order and sorted into separate columns the axial, tangential, and oblique modes. A speed of sound of 1128.6 ft per second was used instead of the 1130 used elsewhere in this book. The spectrum is covered up to 283 Hz which, as we shall see later in this chapter, is enough to explore the region in which most colorations appear.

It must be emphasized that Table 6-3 does not include all the modal frequencies below 283 Hz. If 6 or more integers were used, other frequencies would be interspersed between those shown. However, Table 6-3 is at least a minimum modal density, and other modes would only improve the acoustics of the room, except, possibly, for coincidences.

To get a feel for the distribution of these modal frequencies along a frequency scale, they are plotted in Fig. 6-4. In Fig. 6-4(A) the axial, tangential, and oblique modal frequencies are plotted on separate scales. In Fig. 6-4(B) all are combined on one scale. We note that the favored room dimensional ratio chosen for this room (1.00:1.60:2.33) was a wise one, for there are no serious coincidences (pile-ups) of modes and spacings are reasonably well distributed.

The overall frequency response of the room below 200 Hz is the combined effect of all these resonances. The axial modes are stronger than the tangential modes, and the tangential modes are

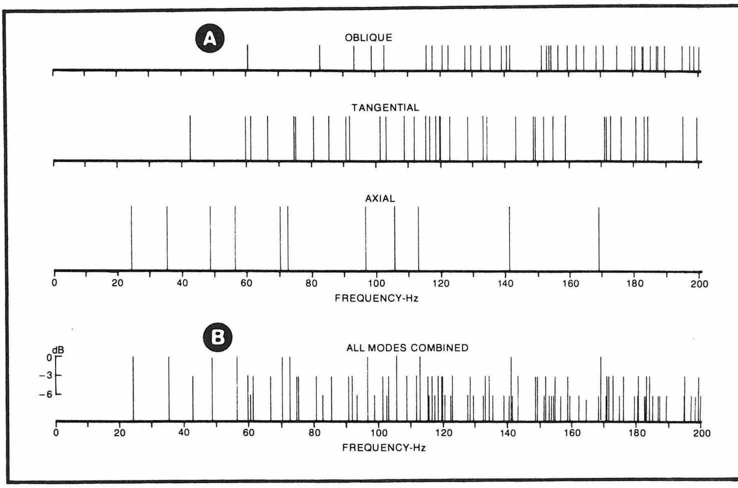

Fig. 6-4. A plot of the modal frequencies for a room 23.3 × 16 × 10 ft taken from Table 6-3. (A) individual plots for oblique modes at the top, tangential modes in the middle, and axial modes below. (B) Combining oblique, tangential, and axial modes. The length of the lines represent the relative strength of the three types of modes.

stronger than the oblique modes. From theoretical considerations[1] the tangential modes are shown to have only ½ the energy and the oblique modes only ¼ the energy of the axial modes for a given pressure amplitude. Taking the axial mode level as the reference, we can plot tangential modes 3 dB down and oblique modes 6 dB down. This explains the different lengths of mode lines in Fig. 6-4. It should be understood, however, that the amount and location of sound absorbing material in the room affects mode intensity, hence in a real room these would vary considerably. For example, the 1,0,0 mode and the 0,1,0 mode involve only the N-S and E-W pairs of wall surfaces. The treatment on the floor and ceiling does not come into play. Conversely, the 0, 0, 1 mode involves only floor and ceiling.

Table 6-3 and Fig. 6-4 apply to the 23.3 × 16 × 10 ft room which is well proportioned according to dimensional ratio C of Table 7-2 and having a volume of 3,728 cu ft, acceptable for a good listening room. Let us now look at another less desirable room having dimensions of 10 × 10 × 8 ft (volume 800 cu ft). Table 6-4 is the printout of modal frequencies for this smaller room, again through the courtesy of Charles Nairn, and Fig. 6-5 is a plot of the frequencies below 200 Hz. The lowest frequency has moved up from 24 Hz to 56 Hz, which means limited low frequency response

L W H	Ax	Tn	Ob
0 1 0	56.43		
1 0 0	56.43		
0 0 1	70.54		
1 1 0		79.80	
0 1 1		90.33	
1 0 1		90.33	
1 1 1			106.51
0 2 0	112.86		
2 0 0	112.86		
1 2 0		126.18	
2 1 0		126.18	
0 2 1		133.09	
2 0 1		133.09	
0 0 2	141.08		
1 2 1			144.56
2 1 1			144.56
0 1 2		151.94	
1 0 2		151.94	
2 2 0		159.61	
1 1 2			162.08
0 3 0	169.29		
3 0 0	169.29		
2 2 1			174.50
1 3 0		178.45	
3 1 0		178.45	
0 2 2		180.67	
2 0 2		180.67	
0 3 1		183.40	
3 0 1		183.40	
1 2 2			189.27
2 1 2			189.27
1 3 1			191.88
3 1 1			191.88
2 3 0		203.46	
3 2 0		203.46	
0 0 3	211.61		
2 2 2			213.02
2 3 1			215.34
3 2 1			215.34
0 1 3		219.01	
1 0 3		219.01	
0 3 2		220.37	

L W H	Ax	Tn	Ob
3 0 2		220.37	
0 4 0	225.72		
4 0 0	225.72		
1 1 3			226.16
1 3 2			227.48
3 1 2			227.48
1 4 0		232.67	
4 1 0		232.67	
0 4 1		236.49	
4 0 1		236.49	
3 3 0		239.41	
0 2 3		239.83	
2 0 3		239.83	
1 4 1			243.13
4 1 1			243.13
1 2 3			246.38
2 1 3			246.38
2 3 2			247.59
3 2 2			247.59
3 3 1			249.59
2 4 0		252.36	
4 2 0		252.36	
2 4 1			262.04
4 2 1			262.04
2 2 3			265.06
0 4 2		266.18	
4 0 2		266.18	
0 3 3		271.00	
3 0 3		271.00	
1 4 2			272.10
4 1 2			272.10
1 3 3			276.81
3 1 3			276.81
3 3 2			277.89
0 0 4	282.15		
3 4 0		282.15	
4 3 0		282.15	
0 1 4		287.74	
1 0 4		287.74	
2 4 2			289.12
4 2 2			289.12
3 4 1			290.84

L W H	Ax	Tn	Ob
4 3 1			290.84
1 1 4			293.22
2 3 3			293.56
3 2 3			293.56
0 2 4		303.89	
2 0 4		303.89	
1 2 4			309.08
2 1 4			309.08
0 4 3		309.40	
4 0 3		309.40	
1 4 3			314.51
4 1 3			314.51
3 4 2			315.46
4 3 2			315.46
4 4 0		319.22	
3 3 3			319.53
2 2 4			324.17
4 4 1			326.92
0 3 4		329.04	
3 0 4		329.04	
2 4 3			329.35
4 2 3			329.35
1 3 4			333.85
3 1 4			333.85
2 3 4			347.86
3 2 4			347.86
4 4 2			349.00
3 4 3			352.69
4 3 3			352.69
0 4 4		361.33	
4 0 4		361.33	
1 4 4			365.71
4 1 4			365.71
3 3 4			370.04
2 4 4			378.55
4 2 4			378.55
4 4 3			382.99
3 4 4			399.02
4 3 4			399.02
4 4 4			426.04

Data prepared by Charles Nairn

Table 6-4. Modal Frequencies for Poorly Proportioned Room. 10 x 10 x 8 Ft. 1,0,0 to 4,4,4.

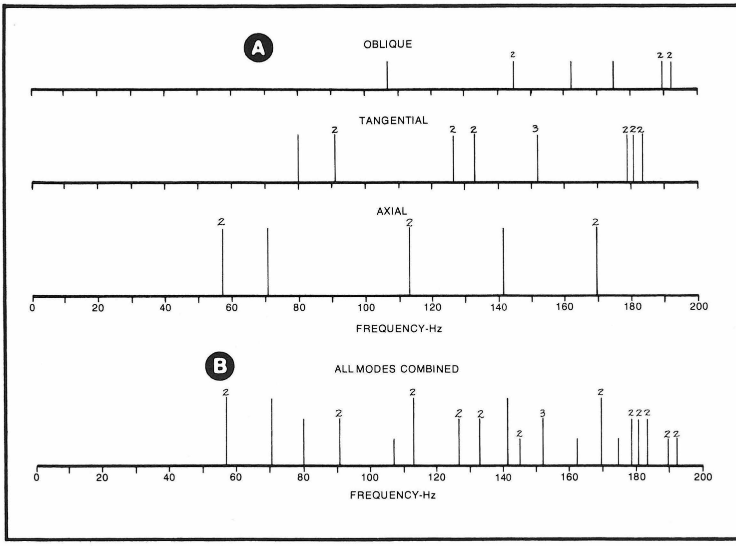

Fig. 6-5. A plot of the modal frequencies of Table 6-4 for a room 10 × 10 × 8 ft which is too small for audio purposes and has unfortunate room proportions. The length and width dimensions being the same results in a great number of degeneracies (coincidences, pile-ups). The number of coincident modes is indicated by a small numeral at that modal frequency.

in the smaller room. Not only are the modes widely spaced, there are numerous degeneracies (coincidences, pile-ups), the number of which is indicated by the small numerals in Fig. 6-5. Thus we see that the small room, poorly proportioned, provides vastly inferior acoustics.

BANDWIDTH OF MODES

In Figs. 6-4 and 6-5 each modal frequency is represented by a slim line. Actually, each has a finite bandwidth, a narrow range of frequencies over which its effect is felt. Resonances in electrical circuits have this same characteristic. The interaction of inductance and capacitance exhibits the common tuning curve. Every time you tune your radio or television set such resonances are employed to provide the necessary selectivity to receive desired and reject undesired signals. This tuning curve is very sharp in low loss (low resistance) circuits and broader in more lossy circuits.

The sharpness of the tuning curve of the acoustical room resonance is also dependent on losses in the "acoustical circuit" in the form of sound absorbents in the room. The "deader" the room (the more absorbent present) the broader the bandwidth of the

resonance curve. Knowing that reverberation time is also dependent on the amount of absorbent present, it follows that the bandwidth of modal resonances is also related to reverberation time according to the following simple empirical equation:[3]

$$\text{Mode Bandwidth} = \frac{2.2}{T_{60}} \qquad \textbf{(6-4)}$$

where T_{60} is the reverberation time of the room in seconds. Bandwidth is defined by the half-power (−3 dB) points on either side of the resonance peak as shown in Fig. 6-6. The modal bandwidth for an average living room or small studio (having a reverberation time of 0.3 second) is 2.2/0.3 = 7.3 Hz. For most small rooms the modal bandwidth falls between 3 and 10 Hz.

With a finite bandwidth for each modal frequency, let us consider in more detail what happens in the open space between the modes of Fig. 6-4(B). As a bandwidth is assigned to each mode, the open space tends to be covered by overlapping skirts of neighboring modes. For example, let us examine in detail the 80-100 Hz region of Fig. 6-4(B) and Table 6-3. There are 8 modes in this frequency span, 1 axial, 4 tangential, and 3 oblique. Assuming a bandwidth of 7 Hz, the overlapping modal tuning curves are something like that of Fig. 6-7. The 3 dB separation between axial and tangential and the 3

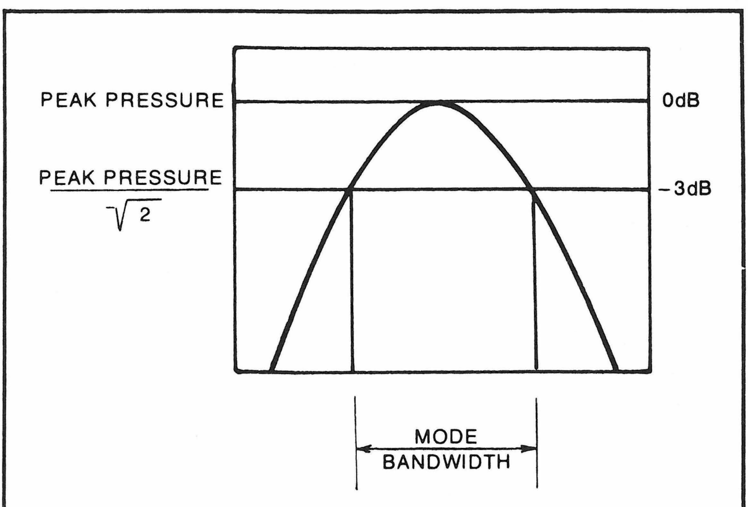

Fig. 6-6. Each mode has a finite bandwidth. The standard way of describing this bandwidth is that width at the −3 dB (half power) points on either side of the resonance frequency.

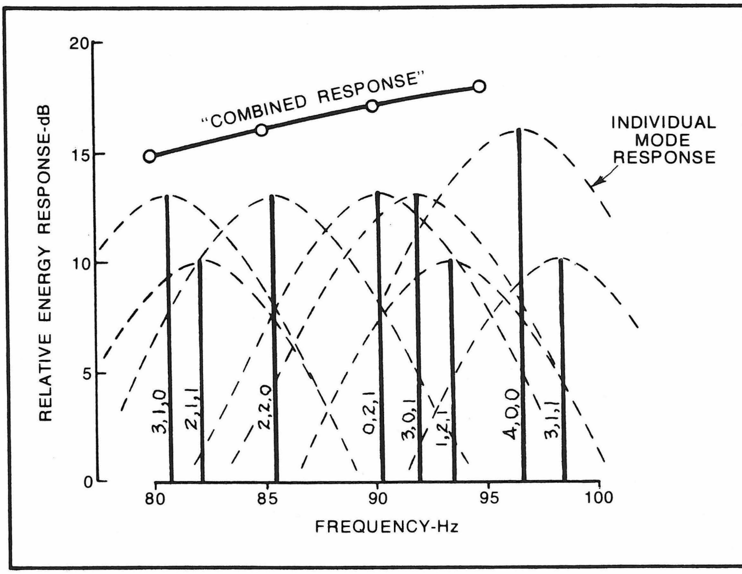

Fig. 6-7. The modal frequencies between 80 and 100 Hz (Table 6-3 and Fig. 6-4) are here plotted with bandwidths of 7 Hz to show the possibility of mutual coupling through overlapping skirts of the resonance curves. The exercise in calculating "combined response" assumes that each mode is at its maximum point at the room position in question, which is not the case. Modal amplitudes and phases vary with room position in a complex way.

dB between tangential and oblique modes is incorporated in this sketch. Considering only the more potent axial modes would, in this 80-100 Hz region, give a wholly inadequate picture of what is going on.

When the plot of Fig. 6-7 was first made, the temptation was irresistible to combine the contributions of all the modes active at a given frequency to get a point on the overall "room response" curve. This was first done at 80 Hz. The 3, 1, 0 mode has a level of 12.8 dB at 80 Hz, the 2, 1, 1 mode a level of 8.5 dB, and the 2, 2, 0 mode a level of 6.5 dB. The others are all below 0 dB and judged to have negligible contributions. Combining these on an energy basis:

$$\begin{aligned}\text{Combined level} &= 10 \log_{10} \left(10^{\frac{12.8}{10}} + 10^{\frac{8.5}{10}} + 10^{\frac{6.5}{10}}\right) \\ &= 14.9 \text{ dB}\end{aligned}$$

This point was plotted as an open circle on the same dB scale. The same calculation was made for 85, 90, and 95 Hz and the resulting points plotted to yield the combined response of Fig. 6-7.

Then it was realized that the above procedure is all wrong. Each modal line in Fig. 6-4 and 6-5 is not a fixed entity for the room, but one whose magnitude varies from zero to maximum depending on position in the room. These are truly standing waves, but their pressure plots vary "all over the map". The first impression of Fig. 6-7 may be that of a sort of average for the room, but this cannot be. Each of the heavy lines of Fig. 6-7 would be some other height between zero and maximum at any given position in the room. To obtain an accurate evaluation of the approach of Fig. 6-7 we must consider the sound pressure variations of each mode.

MODE PRESSURE PLOTS

It is easy to say that the modal pattern of a given room creates a very complex sound field, but to really drive this point home several sketches of sound pressure distributions are included. The one dimension organ pipe of Fig. 5-2 is a starting point which may be compared to the 1,0,0 mode of Fig. 6-8 for a three dimensional room. The pressure is highest near the ends (1.0) and zero along the center of the room. Figure 6-9 shows sound pressure distribution when only the 3,0,0 axial mode is energized. The sound pressure contours in this case are straight lines as shown in Fig. 6-10.

Three dimensional sketches of sound pressure distribution throughout a room become difficult, but Fig. 6-11 is an attempt for

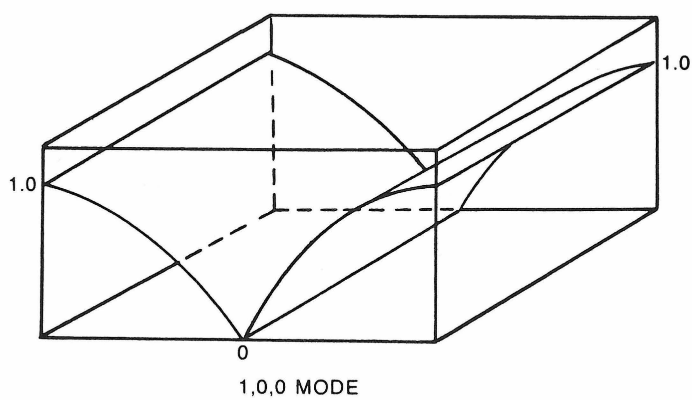

Fig. 6-8. A graphical representation of the sound pressure distribution of the 1,0,0 axial mode of a room. The sound pressure is zero along a line at the center of the room and maximum at the ends of the room. This is comparable to f_1 of the organ pipe of Fig. 5-2.

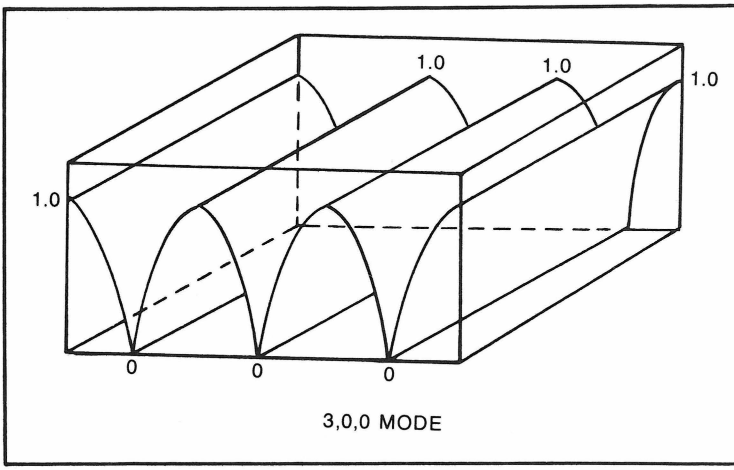

Fig. 6-9. Representation of the sound pressure distribution of the 3,0,0 axial mode of a room.

the 2, 1, 0 tangential mode. We see sound pressure "piled up" in each corner of the room with two more "piles" at the center edges. This is more graphically portrayed in Fig. 6-12 in which the pressure contour lines are drawn. The broken lines crisscrossing the room

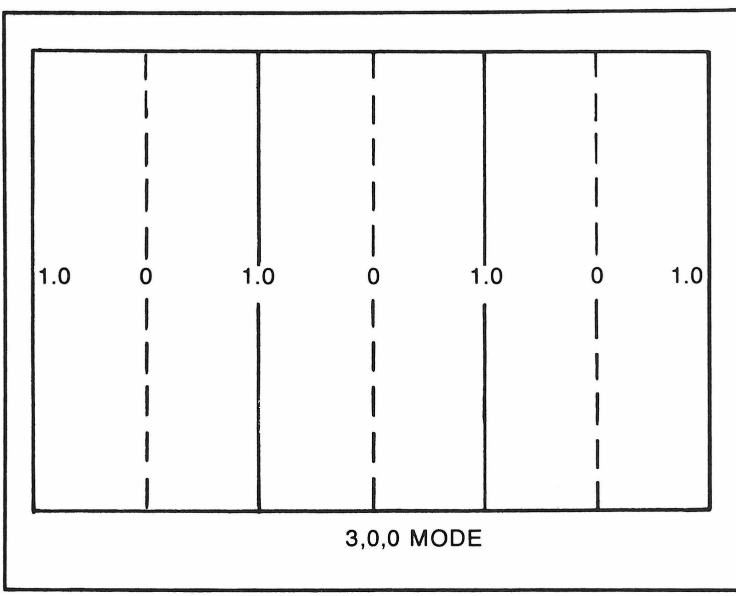

Fig. 6-10. Sound pressure contours on a section through a rectangular room for the 3,0,0 axial mode.

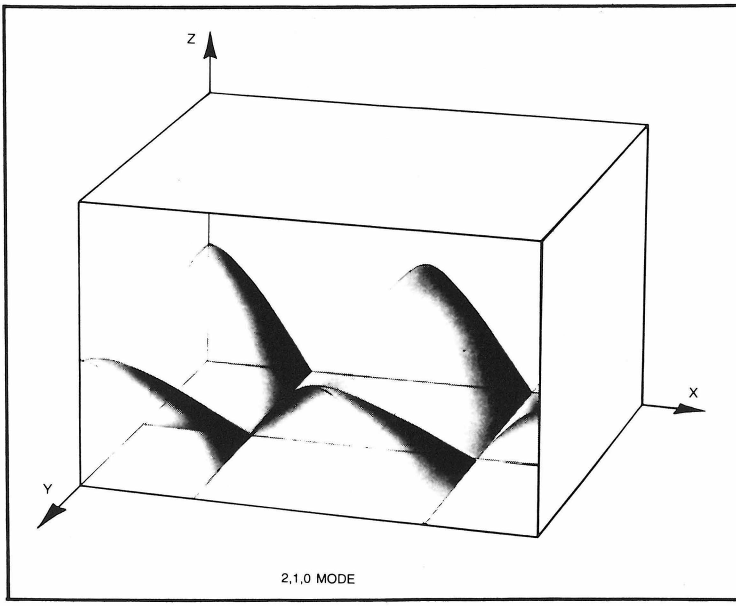

Fig. 6-11. Three dimensional representation of the sound pressure distribution in a rectangular room for the tangential mode 2,1,0. (Brüel & Kjaer Instruments, Inc.[8])

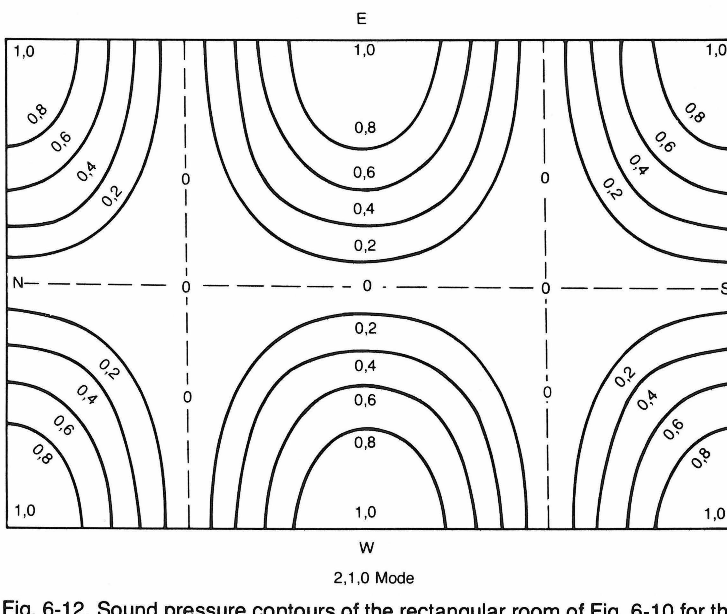

Fig. 6-12. Sound pressure contours of the rectangular room of Fig. 6-10 for the tangential mode 2,1,0. (Brüel & Kjaer Instruments, Inc.[9])

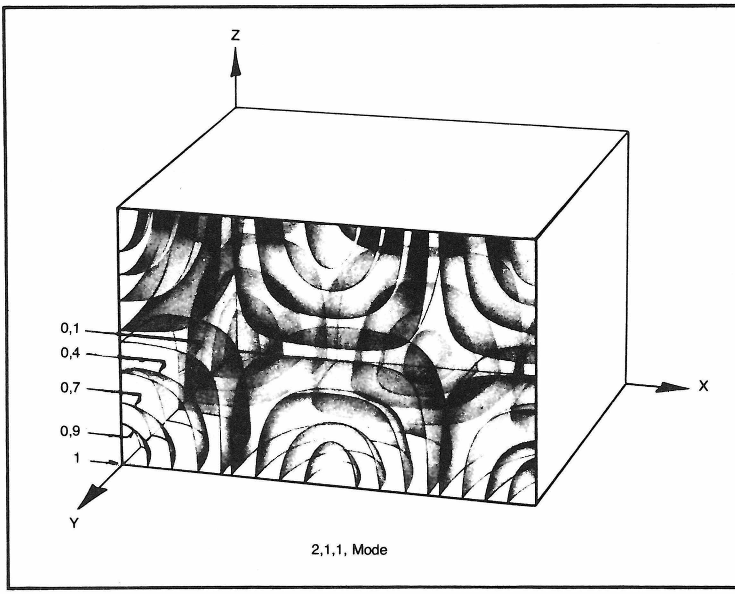

Fig. 6-13. Sound pressure distribution in a rectangular room for the oblique mode 2,1,1. (Brüel & Kjaer Instruments, Inc.[8])

between the "piles" of sound mark the zero pressure regions. The sound pressure distribution for the oblique mode 2, 1, 1 is shown in Fig. 6-13.

Imagine how complicated the sound pressure pattern would be if all the modes of Fig. 6-4 are concurrently or sequentially excited by voice or music energy chasing up and down the spectrum with constantly shifting intensity. The plot of Fig. 6-12 shows pressure maxima in the corners of the room. These maxima always appear in room corners for all modes. If you want to excite all modes, place the sound source in a corner. Conversely, if you wish to measure all modes, a corner is the place to locate the microphone.

AXIAL MODE DOMINANCE

What is the relative importance of axial, tangential, and oblique modes? We have seen that the tangential and oblique modes are 3 and 6 dB, respectively, below the axial modes. Is this separation enough to justify neglecting them in room design and concentrating attention on the axial modes alone? This procedure is widely used in the more casual studio design. If careful attention is given to the selection of room proportions so that the more dominant axial modes are distributed as equitably as possible, then the tangential

and oblique modes can only help by filling in between. Computing all resonances (like Table 6-3) from Eq. 6-1 can be quite a chore when room proportions are being frequently changed (on paper) during the early stages of designing a sound-sensitive room. Calculators and computers programmed for calculating and printing out all modes can be of great assistance to the designer, giving him a powerful tool in room design.

MODE SPACING AND COLORATION

Colorations largely determine the quality of sound for a small studio or listening room. The big task for us, then, is to determine which, if any, of the hundreds of modal frequencies in a room are likely to give us trouble.

It turns out that the spacing of the modal frequencies is a very important factor. Above, say, 300 Hz, the modal frequencies of a small room are so close together that they tend to merge helpfully and harmlessly. At the lower audible frequencies, below about 300 Hz, their separation is greater and it is in this region that problems can arise.

A good question would be, "How close together must adjacent modal frequencies be to avoid problems of coloration?" Gilford[4] states his opinion that an axial mode separated more than 20 Hz from the next axial mode will tend to be isolated acoustically. It will tend not to be excited through coupling due to overlapping skirts but will rather act independently. In this isolated state it can respond to a component of the signal near its own frequency and give this component an unreasonably large resonant boost (which is the very definition of coloration).

Another criterion for mode spacing has been suggested by Bonello[2,3] who considers all three types of modes, not axial modes alone. He states that it is desirable to have all modal frequencies in a critical band at least 5% of their frequency apart. For example, one modal frequency at 20 Hz and another at 21 Hz would be barely acceptable. However, a similar 1 Hz spacing would not be acceptable at 40 Hz (5% of 40 Hz is 2 Hz). Thus we see that Gilford's concern was primarily how far apart axial modes must be spaced to avoid problems resulting from independent and uncoupled modal action. Bonello's concern has to do with separations to avoid degeneracy (coincident) effects.

Zero spacings between modal frequencies are a common source of coloration. Zero spacing means that two modal frequen-

cies are coincident (called a degeneracy by acousticians) and they would tend to overemphasize signal components at that frequency.

EXPERIMENTS WITH COLORATIONS

Any ear can be offended by colorations caused by isolated modes but even a critical and trained ear needs some instrumental assistance in identifying and evaluating such colorations. The BBC Research Department made an interesting study.[4] Observers listened to persons speaking at a microphone in the studio under investigation, the voices being reproduced in another room over a high quality system. Observers' judgments were aided by a selective amplifier which amplified a narrow frequency band (10 Hz) to a level about 25 dB above the rest of the spectrum. The output was mixed in small porportions with the original signal to the loudspeaker, the proportions being adjusted until it is barely perceptible as a contribution to the whole output. Any colorations were then made clearly audible when the selective amplifier was tuned to the appropriate frequency.

In most studios tested this way, and we can assume that they were well designed, only one or two obvious colorations were found in each. Figure 6-14 is a plot of 61 male voice colorations observed

Fig. 6-14. A plot of 61 male voice colorations observed over a period of two years in BBC studios. Most fall in the 100-175 Hz region. Female voice colorations occur between 200 and 300 Hz. (After Gilford[4]).

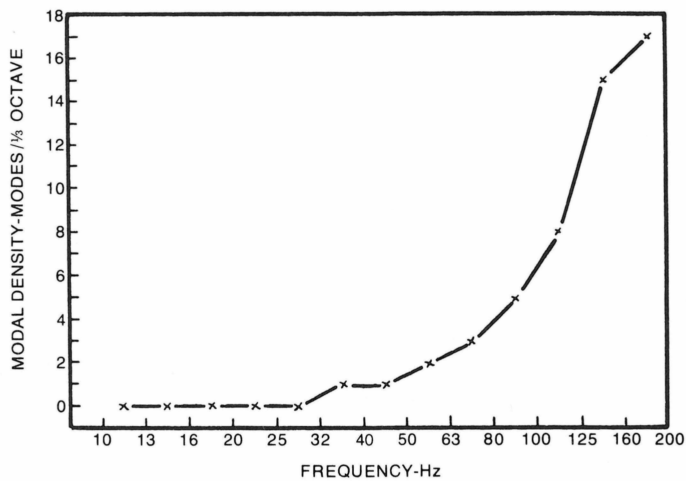

Fig. 6-15. A plot showing the number of modes in 1/3 octave bands for the 15.4 × 12.8 × 10 ft room. The graph climbs steadily upward with no downward anomalies hence the room meets the Bonello criterion.

over a period of 2 years. Most fall between 100 and 175 Hz. Female voice colorations occur most frequently between 200 and 300 Hz.

THE BONELLO CRITERION

Proof that modal resonances in rooms are an international problem, this method of evaluating their effect comes from Buenos Aires. Bonello[2,3] suggests a novel method of determining, by computer, the acoustical desirability of the proportions of rectangular rooms. He divides the low end of the audible spectrum into bands ⅓ octave wide and considers the number of modes in each band below 200 Hz. The ⅓ octave bands are chosen because they approximate the critical bands of the human ear.

To meet Bonello's criterion each ⅓ octave should have more modes than the preceding one, or at least the same number. Modal coincidences are not tolerated unless at least 5 modes are in that band.

How does a 15.4 × 12.8 × 10 ft room qualify by this criterion? Figure 6-15 shows that it passes this test with flying colors. The graph climbs steadily upward with no downward anomalies. The horizontal section at 40 Hz is allowed. The advantage of Bonello's

plan is that it is well adapted to computer calculation and print-out.

MODAL DENSITY

We note in Fig. 6-4 the tendency toward greater modal density with increase in frequency. In the 20 Hz spread between 40 and 60 Hz only 4 modal frequencies are counted. Between 80 and 100 Hz there are 8 modes. Between 160 and 180 Hz are 11 modes. Even in this very limited low frequency range below 200 Hz we see modal density increasing with frequency. In Fig. 6-16 we see that at somewhat higher frequencies the rate of increase dramatically rises. In fact, above about 300 Hz or so the mode spacing is so small that the room response smooths markedly with frequency.

FREQUENCY REGIONS

The audible spectrum is very wide when viewed in terms of wavelength. At 16 Hz, considered the low frequency limit of the average human ear, the wavelength is 1130/16 = 70.6 ft. At the upper extreme of hearing, say 20,000 Hz, the wavelength is only 1130/20,000 = 0.0565 ft or about 11/16th of an inch. The behavior

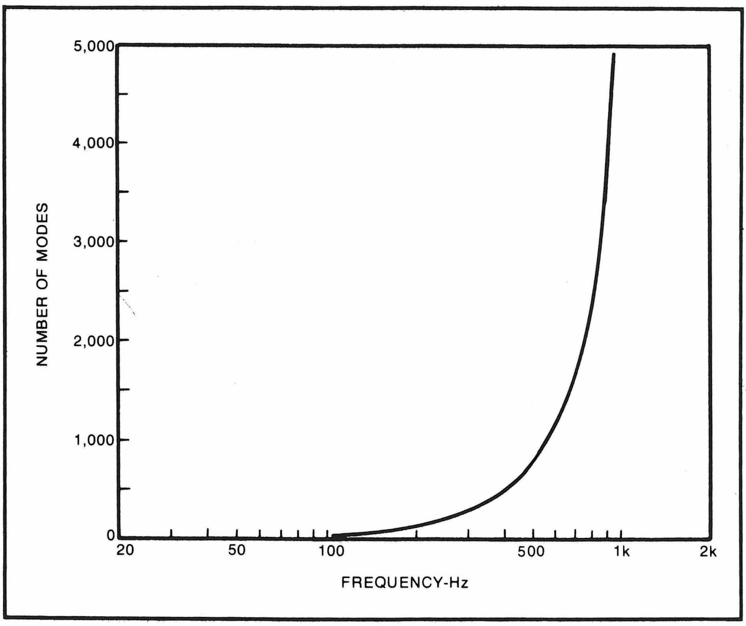

Fig. 6-16. The number of modal frequencies increases dramatically with frequency.

of sound is affected greatly by the wavelength of the sound in comparison to the size of objects encountered. In a room, sound of 11/16 inch wavelength is scattered (diffused) significantly by a wall irregularity of a few inches. The effect of the same irregularity on sound of 70 ft wavelength would be insignificantly small. The heart of the acoustical problem in the usual audio room is that no single analytical approach can cover sound of such a wide range of wavelengths.

In considering the acoustics of small rooms, let us arbitrarily divide the audible spectrum into four regions, A, B, C, and D. Region A is the very low frequency region below a frequency of 1130/2L or 565/L where L is the longest dimension of the room. Below the frequency of this lowest axial mode there is no resonant rise at all for sound in the room. This does not mean that such very low frequency sound cannot exist in the room, only that it is not boosted by room resonances because there are none in that region.

Region B is that region we have studied in detail in which the dimensions of the room are comparable to the wavelength of sound being considered. It is bounded on the low frequency end by the lowest axial mode, 565/L. The upper boundary is not definite but an approximation is given by what has been called the "cutoff" or "crossover" frequency given by the equation:[5]

$$F = 11{,}250 \sqrt{\frac{T_{60}}{V}} \tag{6-4}$$

where

F = cutoff or crossover frequency, Hz.
T_{60} = reverberation time of the room, seconds.
V = volume of the room, cu ft.

In the 10 × 16 × 23.3 ft room of Table 6-3 and Fig. 6-4, the volume is 3728 cu ft. For a reverberation time of 0.5 second the crossover frequency is 130 Hz. A smaller room of dimensions 10 × 12.8 × 15.4 ft having a volume of 1971 cu ft and a reverberation time of 0.5 second yields a cutoff frequency of 179 Hz. A glance at Fig. 6-4 reveals that the modal density is greater above and lower below the cutoff frequency of 130 Hz. Nothing happens suddenly at this frequency it only designates the approximate location of a gradual transition region below which room resonance dominate.

Region D covers the higher audible frequencies for which the wavelengths are short enough for geometric acoustics to apply.

Specular reflections (angle of incidence equals angle of reflection) and the sound ray approach to acoustics prevail. In this region statistical approaches are generally possible.

Retracing our steps, Region C is a transition region between Region B, in which wave acoustics must be used, and Region D in which ray acoustics are valid. It is bounded on the low frequency end approximately by the cutoff frequency F of Eq. 6-4 and on the high end approximately by 4F. It is a difficult region dominated by wavelengths often too long for ray acoustics and too short for wave acoustics.

For the 10 × 16 × 23.3 ft room, below 565/23.3 = 24.2 Hz is Region A in which there is no resonant boost for sound. Between 24.2 Hz and 130 Hz (Eq. 6-4) the wave acoustical approach of modal resonances is essential. Between 130 Hz and (4) (130) = 520 Hz is the transition Region C. Above about 520 Hz the modal density is very high, statistical conditions generally prevail, and the simpler geometrical acoustics can be used. Room size determines how the audible spectrum must be divided for acoustical analysis. Very small rooms, with too few modal resonances spaced too far apart, are characterized by domination of a great stretch of the audible spectrum by modal resonances. This is the "small studio problem" in a nutshell.

SIMPLIFIED AXIAL MODE ANALYSIS

Let us apply what we have learned about axial modes to a specific rectangular listening room or studio. The dimensions of our specimen room are 28 × 16 × 10 ft. The 28 foot length resonates at 565/28 = 20.2 Hz, the two side walls 16 ft apart resonate at 565/16 = 35.3 Hz, and the floor-ceiling combination resonates at 565/10 = 56.5 Hz. These three axial resonances and the train of multiples for each are plotted in Fig. 6-17. There are 27 axial resonance frequencies below 300 Hz and, for this exercise, we are neglecting the horde of weaker tangential and oblique modes.

Because most signal colorations are traceable to axial modes, let us examine their spacings in detail. Table 6-5 illustrates a convenient form for this simplified analysis of axial modes. The resonance frequencies from the L, W, and H columns on the left of Table 6-5 are arranged in ascending order. This makes it easy to examine that critical factor, axial mode spacing.

We note that the L-f_7 resonance at 141.3 Hz coincides with the W-f_4 resonance. This means that these two potent axial modes team together to create a potential coloration of sound at that frequency.

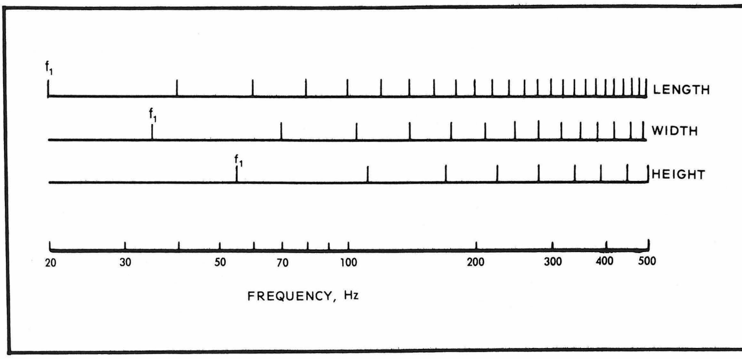

Fig. 6-17. The axial modal frequencies and multiples of the 16 × 28 × 10 ft room.

Table 6-5. Axial Mode Analysis Form.

Room dimensions: Length 28.0 ft
Width 16.0 ft
Height 10.0 ft

	Axial Mode Resonances Hz			Arranged in ascending order	Axial mode spacing Hz
	L	W	H	20.2	15.1
f_1	20.2	35.3	56.5	35.3	5.1
f_2	40.4	70.6	113.0	40.4	16.1
f_3	60.5	105.9	169.5	56.5	4.0
f_4	80.7	141.3	226.0	60.5	10.1
f_5	100.9	176.6	282.5	70.6	10.1
f_6	121.1	211.9	339.0	80.7	20.2
f_7	141.3	247.2		100.9	5.0
f_8	161.4	282.5		105.9	7.1
f_9	181.6	317.8		113.0	8.1
f_{10}	201.8			121.1	20.2
f_{11}	222.0			141.3	0
f_{12}	242.1			141.3	20.1
f_{13}	262.3			161.4	8.1
f_{14}	282.5			169.5	7.1
f_{15}	302.7			176.6	5.0
				181.6	20.2
				201.8	10.1
				211.9	10.1
				222.0	4.0
				226.0	16.1
				242.1	5.1
				247.2	15.1
				262.3	20.2
				282.5	0
				282.5	0
				282.5	20.2
				302.7	

This coincidence is also separated 20 Hz from its neighbors. We also note from Fig. 6-14 that 141.3 Hz is in a frequency range that is especially troublesome. Here, then, is a warning of a potential problem.

At 282.5 Hz we see a "pile-up" of L-f_{14}, W-f_8, and H-f_5 modes which, together, would seem to be an especially troublesome source of colorations. They are also separated from neighbors by 20 Hz. However, looking at the experimentally derived plot of Fig. 6-14, practically no problem, with voice colorations were found at 282 Hz, at least for male voices. The reason for this is the salutary presence of tangential and oblique modes neglected in this study.

With the threat of coloration at 141 Hz, adjustment of dimensions of a proposed room would be a logical attack. If it is an existing room, a Helmholtz resonator, sharply tuned and properly located, is a difficult, but possible solution.

CONTROLLING PROBLEM MODES

The general construction of Helmholtz resonators for normal room treatment is detailed in Chapter 9, but building one with very sharp tuning (high Q) is more demanding[6]. The flexing of wooden boxes introduces losses which lower the Q. Attaining a truly high Q resonator with sharp tuning the cavity must be made of concrete, ceramic, or other hard, non-yielding material, but fitted with some means of varying the resonance frequency. The resonance frequency of a Coca Cola bottle was measured[7] at 185 Hz and was found to have a bandwidth (−3 dB points) of 0.67 Hz. This yields a Q = 185/0.67 *=276, a very high value. So, if you are fortunate enough to have your mode problem at 185 Hz . . . . !*

It is also important where the Coke bottle (or other Helmholtz absorber) is placed if the goal is to bring a mode or closely spaced group of modes under control. Let us say that the 2,1,0 mode of Fig. 6-12 is causing a voice coloration and that it is necessary to introduce a narrow sliver of absorption at the 2,1,0 frequency. If the *Helmholtz absorber were placed at a pressure node (zero pressure)* it would have, obviously, no effect. Placed at one of the antinodes (pressure peaks) it would have tight interaction with the 2,1,0 mode. Therefore, any corner would be acceptable, as would the pressure peaks on the E or W walls.

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7

Diffusion of Sound

Our goal is to have sound energy over the entire audible frequency range uniformly distributed throughout the enclosure. Such sound diffusion simplifies microphone placement in a studio and assures uniform listening conditions in a listening room. A valuable by-product is that reverberation calculations based on the assumption of completely diffuse conditions give dependable answers. Of even greater significance, however, is the control of coloration of sound originating in a studio or being reproduced in a control room or other listening room.

A highly simplified picture of a nondiffuse condition was seen in the pipe with closed ends of Fig. 5-2. Considering only two parallel walls of a room and only the lowest resonance frequency, f_1, the sound pressure is high near the walls and is zero at the center of the room. This is a real physical effect, not just a theoretical concept. However, the $2f_1$ and the $3f_1$ and other multiples of f_1 that would be excited simultaneously by a complex voice or music signal would tend to obscure the pristine simplicity of the f_1 situation. The restoration of the other four surfaces and their attendant axial modes would further enormously complicate matters.

ROOM PROPORTIONS

If any two or three dimensions of a rectangular room are the same or if dimensions are in multiple relationship to each other, modal frequencies will coincide and there will be an inordinate

Table 7-1. Coincident Room Resonances.

Axial Modes	Height 10 ft	Width 20 ft	Length 30 ft
f_1	56.5	28.3	18.8
$2f_1$	113	56.5	37.7
$3f_1$	169.5	84.8	56.5
$4f_1$	226.0	113.0	75.3
$5f_1$	282.5	141.3	94.2
$6f_1$	339.0	169.5	113.0
$7f_1$	395.5	197.8	131.8
$8f_1$	452.0	226.0	150.7
$9f_1$	508.5	254.3	169.5
$10f_1$	565.0	282.5	188.3
$11f_1$	621.5	310.8	207.2
$12f_1$	678.0	339.0	226.0
$13f_1$	734.5	367.3	244.8

boosting of sound at these frequencies. For example, in a room whose length is 30 ft, width 20 ft, and height 10 ft we see this multiple relationship. The axial modal frequencies of this room are tabulated in Table 7-1. We see that 56.5, 113, 169.5, and 226 Hz are common to all three axes of the room. This room therefore has many coincident frequencies and maximum sound diffusion is not achieved.

Coincident frequencies result in greater average spacing between modal frequencies. Therefore, we have not only greater energy concentrated at the coincident frequencies, but less energy between them. This means a less diffused field. The physicist calls coincident frequencies "degeneracies", a departure from the diffuse state.

There are preferred ratios of room dimensions (derived mathematically from wave acoustics), the goal of which is to distribute the modal frequencies in an optimum manner. The need for proper proportioning of an audio room is illustrated by the 10×20×30 ft room of Table 7-1. The lowest axial mode for the 10 ft height is $f_1 = 1130/(2)(10) = 56.5$ Hz. The floor/ceiling mode is also resonant at 2 times, 3 times, 4 times 56.5 Hz. In a similar way, axial mode frequencies are figured for width and length dimensions. We note that 56.5 Hz. appears in each of the three columns. This means that any distributed sound in the room, such as voice or music, receives a boost as a result of a peak in room response at 56.5 Hz, a "triple whammy" as it were. The same thing happens at

113 Hz, 169.5 Hz, 226 Hz, etc. There are similar modal coincidences, or pile-ups, on up the frequency scale but we are interested only in frequencies below 300 or 400 Hz where the modal density is low and coloration problems thrive. We conclude that the ratio of room dimensions of this room, 1:2:3, is not the best. For the best ratios we must search elsewhere.

Away back in 1946 Bolt published a paper[1] that specified a

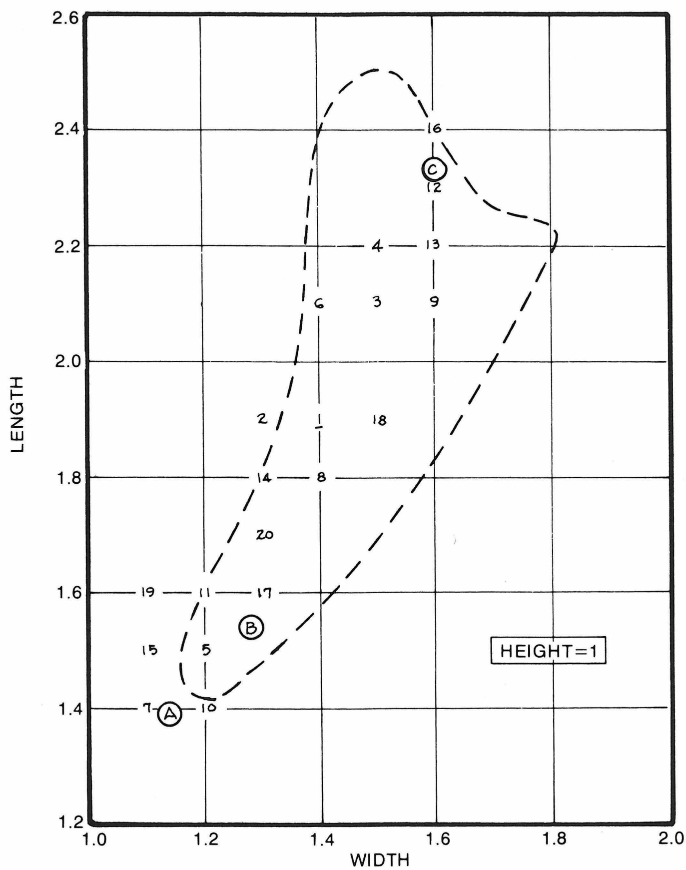

Fig. 7-1. A chart of favorable room dimensional ratios to achieve uniform distribution of modal frequencies of a room. The broken line encloses the so-called "Bolt area" of Reference 1. The encircled letters A, B, and C are room proportions Sepmeyer lists as favorable in Reference 2. The numbers 1 through 20 are Louden's 20 "best quality" dimensional ratios in descending order of quality[3]. All three studies show a high degree of consistency in spite of differing criteria employed.

Table 7-2. Sepmeyer's Room Dimension Ratios.[2] Reference 7-2

	A	B	C
Height	1.00	1.00	1.00
Width	1.14	1.28	1.60
Length	1.39	1.54	2.33

general range of room dimensional ratios which promised to give a satisfactory distribution of modal frequencies. This is shown in Fig. 7-1 as a broken line defining an area of "good" ratios. Room proportions falling within this area should reduce (but not necessarily eliminate) coincidences and spread out the modal frequencies more or less uniformly. Conversely, ratios falling outside the area are, by Bolt's criteria, less desirable.

In 1965 Sepmeyer published the results of another mathematical study of normal mode distribution.[2] Three ratios growing out of this study are shown in Table 7-2 and are plotted in Fig. 7-1 as the three encircled letters, A, B, and C. B and C fall within Bolt's area,

Table 7-3. Preferred Room Dimensions.

(Using Sepmeyer's Ratios)	A	B	C
		Ceiling Height-8 ft	
Height	8.00 ft	8.00 ft	8.00 ft
Width	9.12	10.24	12.80
Length	11.12	12.32	18.64
Volume, ft^3	(811)	(1009)	(1909)
		Ceiling Height-10 ft	
Height	10.00	10.00	10.00
Width	11.40	12.80	16.00
Length	13.90	15.40	23.30
Volume, ft^3	(1585)	(1971)	(3728)
		Ceiling Height-12 ft	
Height	12.00	12.00	12.00
Width	13.68	15.36	19.20
Length	16.68	18.48	27.96
Volume, ft^3	(2738)	(3406)	(6442)
		Ceiling Height-14 ft	
Height	14.00	14.00	14.00
Width	15.96	17.92	22.40
Length	19.46	21.56	32.62
Volume, ft^3	(4348)	(5409)	(10230)

but A is slightly outside. Sepmeyer's three ratios are translated into specific room dimensions for ceiling heights from 8 ft to 14 ft in Table 7-3. Room volumes for each combination of dimensions are shown because if a room of a given size (volume) is required and one dimension is fixed, some ratios are inapplicable. It should be remembered that length, width, and height are labels that hold no significance for sound waves. Standing a long room on end, if there is unlimited ceiling height, but meeting ratio and volume criteria, is quite acceptable.

Coming down to relatively recent times it is instructive to examine Louden's findings published in 1971.[3] Louden listed 125 room dimensional ratios arranged in order of decreasing "quality". The first (and best) 20 are listed in Table 7-4 and are also plotted in Fig. 7-1. Only 5 of the 20 fall outside Bolt's area. All three studies show a considerable affinity for each other as Fig. 7-1 is studied. Any of the three may be followed to establish tentative initial proportions in an audio room design. In any event, the tentative dimensions should be subjected to a detailed study of modal frequency distribution, not only of axial modes, but of tangential and oblique modes as well, as outlined in Chapter 6.

Having favorable room proportions by no means assures good acoustics, but it is the logical and most favorable start in the quest.

Table 7-4. Louden's Room Dimension Ratios[3]. Reference 7-3.

Order of Quality	1 : X	: Y
1	1.9	1.4
2	1.9	1.3
3	1.5	2.1
4	1.5	2.2
5	1.2	1.5
6	1.4	2.1
7	1.1	1.4
8	1.8	1.4
9	1.6	2.1
10	1.2	1.4
11	1.6	1.2
12	1.6	2.3
13	1.6	2.2
14	1.8	1.3
15	1.1	1.5
16	1.6	2.4
17	1.6	1.3
18	1.9	1.5
19	1.1	1.6
20	1.3	1.7

Nor does an existing recording studio or listening room whose proportions lie outside the acceptable area mean unacceptable acoustics, but it does mean that more care is required in salvaging a less than optimum situation.

FLUTTER ECHOES

When reflections are periodic, or near-periodic, they are especially disturbing and have been given the name, "flutter echoes."[4,5] They are the result of repeated reflections from plane parallel surfaces. Flutter echoes can be easily observed in corridors and long rooms in which the floor, ceiling, and side walls are absorbent but the ends reflective. They can also occur with less extreme room geometry and treatment. Their audibility is usually associated with certain positions of source and listener.

Flutter echoes are especially disturbing because our hearing is very sensitive to periodic repetitions of sound. If the time delay between successive pulses of the flutter is very short, it may be perceived as a tonal coloration of reverberation, speech, or music. If the time delay between pulses is greater than 30 to 50 milliseconds (outside the Haas fusion zone), the periodic structure itself becomes audible as flutter.

Flutter echoes are evidence of poor attention to diffusion of sound in a room. Concentration of required sound absorbing material on one or two surfaces may result in production of flutter or other sound deficiencies. Never should two opposing, parallel walls of a room be more highly reflective (less absorbent) than the other two pairs of surfaces. Flutter can almost always be eliminated by a few modest patches of absorbing material on one or both of the offending surfaces.

NONRECTANGULAR ROOMS

If a studio or listening room is built with one or two of its walls off square, will this eliminate the standing waves and all their problems? Certainly it would tend toward eliminating flutter and would also tend to diffuse sound in the upper part of the audible spectrum where sound rays have meaning. Room resonance effects at the bass frequencies, however, would still be present as their identity is basically associated with volume more than shape.

The acoustical benefit to be derived in the use of nonrectangular shapes in audio rooms is rather controversial. Gilford states, ". . . slanting the walls to avoid parallel surfaces . . . does not remove colorations; it only makes them more difficult to predict."[6]

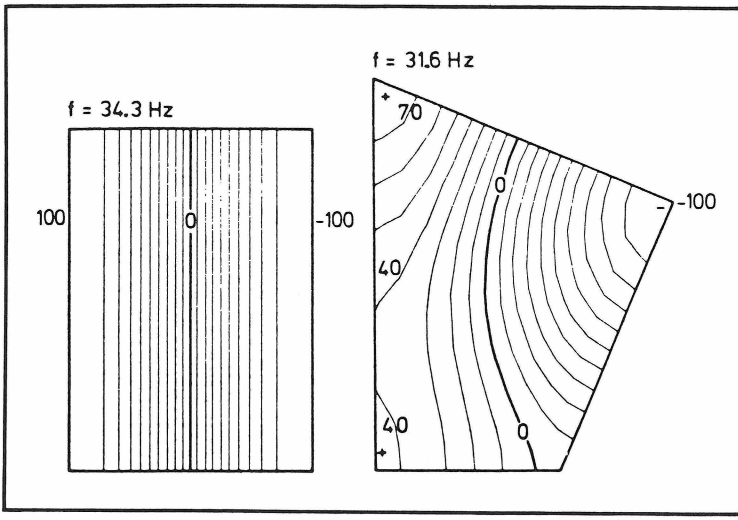

Fig. 7-2. Comparison of the modal pattern for a 5 × 7 meter two-dimensional room and a nonrectangular room of the same area. This sound field of the 1,0 mode is distorted in the nonrectangular room and the frequency of the standing wave is shifted slightly. (From van Nieuwland and Weber, Noise Control Engineering.[7])

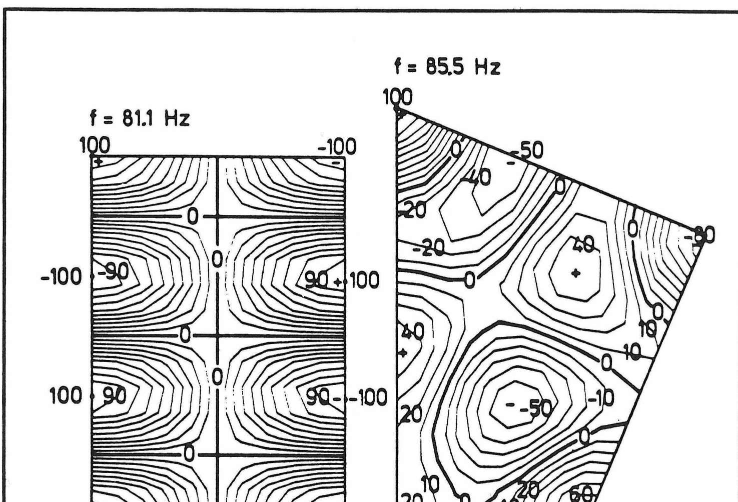

Fig. 7-3. The 1,3 mode for the 5 × 7 meter room of Fig. 7-2 compared to a nonrectangular room of the same area. The sound field is distorted and the frequency is shifted. (From van Nieuwland and Weber, Noise Control Engineering[7])

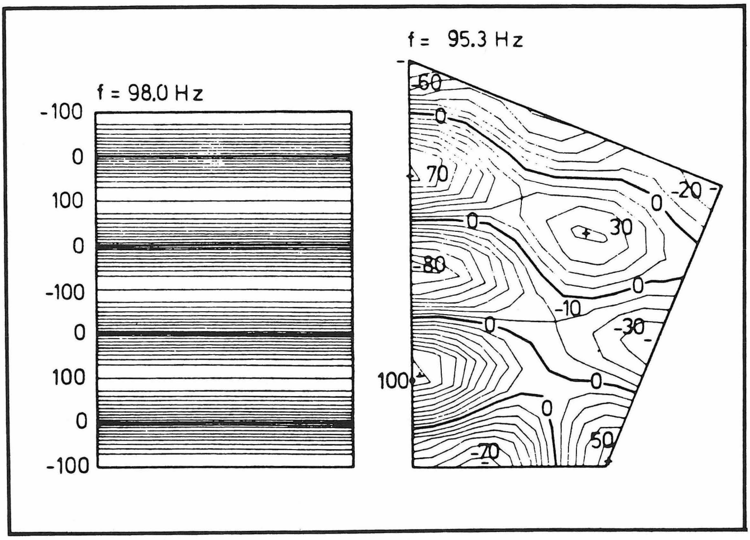

Fig. 7-4. The 0,4 mode of the 5 × 7 meter two-dimensional room of Figs. 7-2 and 7-3. The high distortion in the nonrectangular room is accompanied by a shift in standing wave frequency. (From van Nieuwland and Weber, Noise Control Engineering[7])

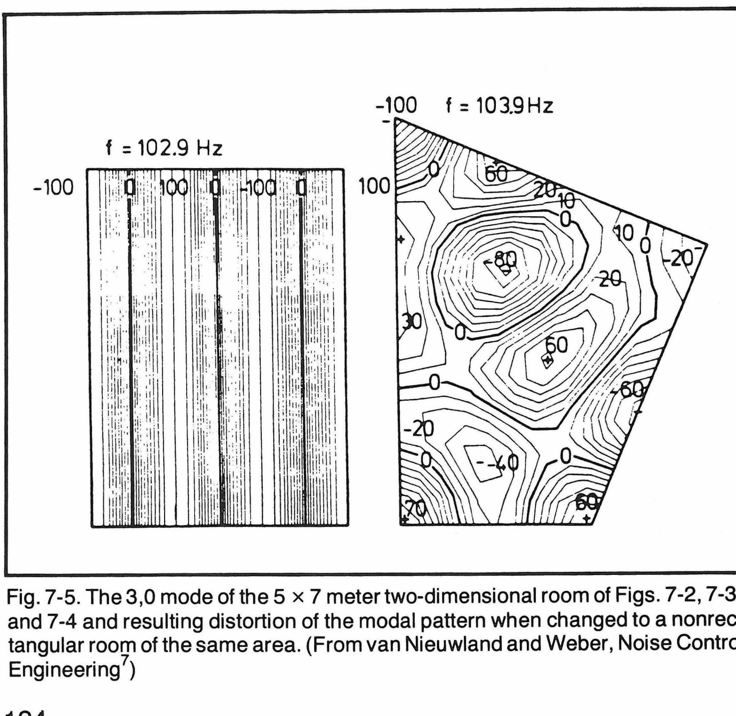

Fig. 7-5. The 3,0 mode of the 5 × 7 meter two-dimensional room of Figs. 7-2, 7-3, and 7-4 and resulting distortion of the modal pattern when changed to a nonrectangular room of the same area. (From van Nieuwland and Weber, Noise Control Engineering[7])

Massive trapezoidal shaped walls, commonly used as the outer shells of recording studio control rooms, guarantee asymmetrical low frequency sound fields even though it is generally conceded that symmetry with the control position is desirable.

Computer studies with the finite element approach have revealed in minute detail what happens to a low frequency sound field in nonrectangular rooms.[7] The results of a study by van Nieuwland and Weber at Philips Research Laboratories, The Netherlands, using this method are given in Figs. 7-2, 7-3, 7-4, and 7-5. Highly contorted sound fields are shown, as expected for the nonrectangular case, for modes 1,0, 1,3, 0,4, and 3,0. A shift in frequency of the standing wave from that of the rectangular room is indicated: −8.6%, −5.4%, −2.8%, and +1% in the four cases illustrated. This would tend to support the common statement that splaying of walls helps slightly in breaking up degeneracies, but shifts of 5% or more are needed to avoid the effects of degeneracies. However, the proportions of a rectangular room can be selected to eliminate, or at

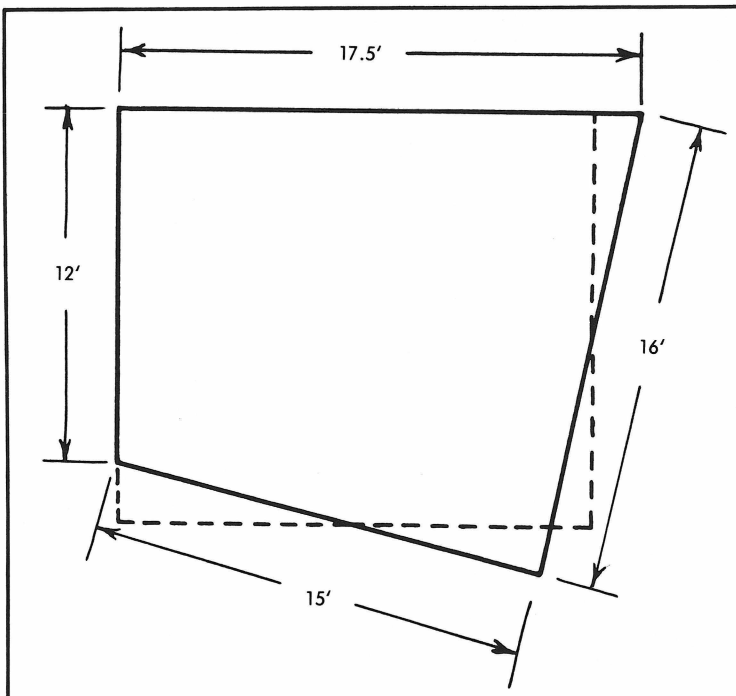

Fig. 7-6. Splaying the walls does not eliminate room resonances at the bass frequencies, it only makes the sound field more complex and unpredictable.

least greatly reduce, degeneracies, while in the case of a nonrectangular room such a prior examination of degeneracies is completely impractical. Just making the sound field asymmetrical by splaying walls only introduces unpredictability in listening room and studio situations.

If the decision is made to splay walls in an audio room, say 5%, a reasonable approximation would be to analyze the equivalent rectangular room having the same volume. Figure 7-6 pictures a room with two splayed walls with wall lengths of 12, 17.5, 16, and 15 ft. The equivalent rectangular walls may then be drawn through the midpoints of each splayed wall. This reduces the room to an equivalent 16 × 14 ft rectangular room for axial mode analysis. It is obvious that the nonrectangular walls of Fig. 7-6 would disturb two of the three sets of axial modes, but would leave the floor-ceiling set untouched.

CONCAVE SURFACES

A concave surface such as that in Fig. 7-7(A) tends to focus sound energy and consequently should be avoided because focusing is just the opposite of the diffusion we are seeking. The radius of curvature determines the focal distance; the flatter the concave surface the greater the distance at which sound is concentrated. Such surfaces often cause problems in microphone placement. Concave surfaces might produce some awe-inspiring effects in a whispering gallery where you can hear a pin drop 100 feet away, but they are to be avoided in listening rooms and small studios.

CONVEX SURFACES: THE POLY

One of the most effective diffusing elements, and one relatively easy to construct, is the polycylindrical diffuser (poly) which presents a convex section of a cylinder. Three things can happen to sound falling on such a cylindrical surface made of plywood or hardboard; (1) the sound can be reflected and thereby dispersed as in Fig. 7-7(B), (2) the sound can be absorbed, or (3) the sound can be reradiated. Such cylindrical elements lend themselves to serving as absorbers in the low frequency range where absorption and diffusion are so badly needed in small rooms. The reradiated portion, because of the diaphragm action, is radiated almost equally throughout an angle of roughly 120° as shown in Fig. 7-8(A). A similar flat element reradiates sound in a much narrower angle, about 20°. Therefore, favorable reflection, absorption, and reradiation characteristics favor the use of the cylindrical surface. Some

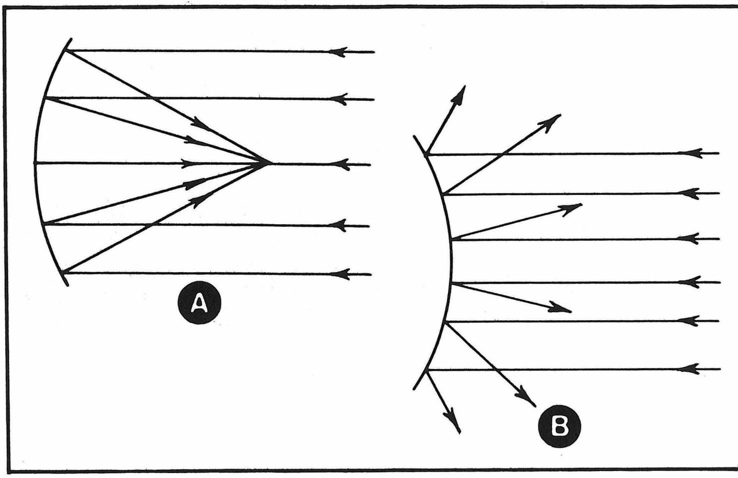

Fig. 7-7. Concave surfaces tend to focus sound, convex surfaces tend to diffuse it. Concave surfaces should be avoided if the goal is to achieve well diffused sound.

very practical polys and their absorption characteristics are presented in Chapter 9. The dimensions of such diffusers are not critical, although to be effective their size must be comparable to the wavelength of the sound being considered. The wavelength of sound at 1000 Hz is a bit over 1 foot, at 100 Hz about 11 feet. A poly element 3 or 4 ft across would be effective at 1000 Hz, much less so at 100 Hz. In general, poly base or chord length of 2 to 6 feet with depths of 6 to 18 inches meet most needs.

It is important that diffusing elements be characterized by randomness. A wall full of polys, all of 2 foot chord and of the same depth, might be beautiful to behold, like some giant washboard, but not very effective as diffusers. The regularity of the structure would cause it to act as a diffraction grating, affecting one particular frequency in a much different way than other frequencies, which is opposite to what the ideal diffuser should do.

Axes of symmetry of the polys on different room surfaces should be mutually perpendicular.

PLANE SURFACES

Geometrical sound diffusing elements made up of two flat surfaces to give a triangular cross section or of three or four flat surfaces to give a polygonal cross section may also be used. In general, their diffusing qualities are inferior to the cylindrical section.

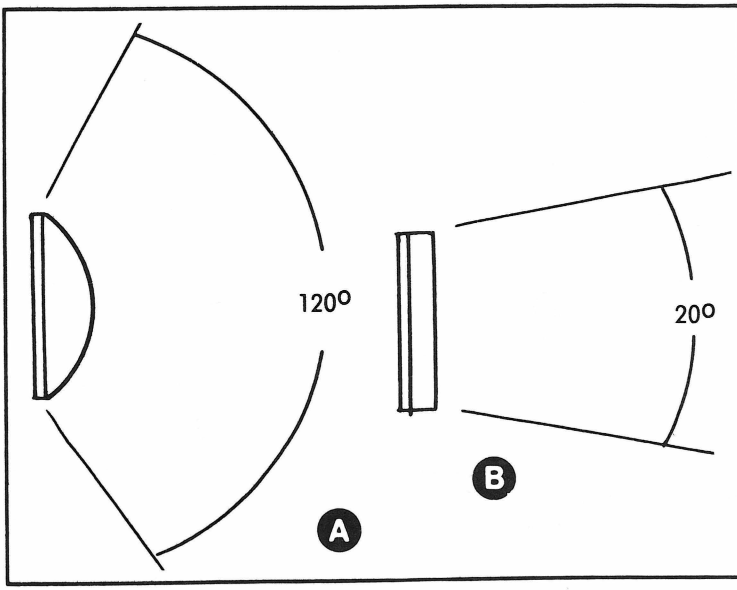

Fig. 7-8. (A) A polycylindrical diffuser reradiates sound energy not absorbed through an angle of about 120°. (B) A similar flat element reradiates sound in a much smaller angle.

DISTRIBUTION OF ABSORBING MATERIALS

It has long been known that the distribution of sound absorbing materials in patches throughout the room has a beneficial effect on the diffusion of sound. In practice, absorbing materials may be applied in geometrical designs to achieve both the desired diffusion and a pleasing esthetic effect.

The acoustical specialists of the British Broadcasting Corporation compared two identical studios having approximately the same amount and types of absorption. One of the studios had diffusers in the form of cuboid protuberances fitted to the walls. Extensive reverberation and listening tests were inconclusive in establishing any acoustical superiority of one over the other. Adding the cuboid diffusers is expensive and of questionable esthetic value, hence the BBC now depends solely upon careful placement of different types of acoustical absorbers for their diffusion.

The secret of the BBC success is a modular approach to studio treatment. They use standard modules of 2 foot width, built to have several different acoustical characteristics (wideband absorbers, bass absorbers, etc.). The walls are completely covered with modules of the various types required, distributed properly (see Fig.

9-27). The ceilings are covered with perforated metal trays concealing bass absorbers to compensate for the high frequency absorption of the carpets. The BBC has achieved adequate diffusion in several hundred studios by careful distribution of the modules of different types.

GROOVY WALLS

Schroeder[8] has injected a new idea into the search for a means of diffusing sound. Present interest and application of his idea is directed primarily to concert halls, but it is quite probable that application to recording studios and other audio rooms is in the offing. In computer science maximum-length codes are applied in generating binary pseudo-random noise signals, the power spectrum of which is absolutely flat. Applying the same mathematics, a diffusing surface is composed of a certain array of reflection coefficients of +1 and −1. The +1 is the usual coefficient of any good reflector. The −1 coefficient is obtained by grooves in the surface $\frac{\lambda}{4}$ deep. In Fig. 7-9 is illustrated a pseudo-random maximum-length code N =15. Sound falling on this type of surface from a given direction is reflected (dispersed) almost uniformly throughout 180°. Theoretically, an infinite expanse of the arrays of Fig. 7-9 is required, but practically, even a single array is an excellent diffuser. There is a certain design frequency by which the $\frac{\lambda}{4}$ and $\frac{\lambda}{2}$ dimensions of the array are determined. Such an array works well as a diffuser over about one octave, half octave above and half octave

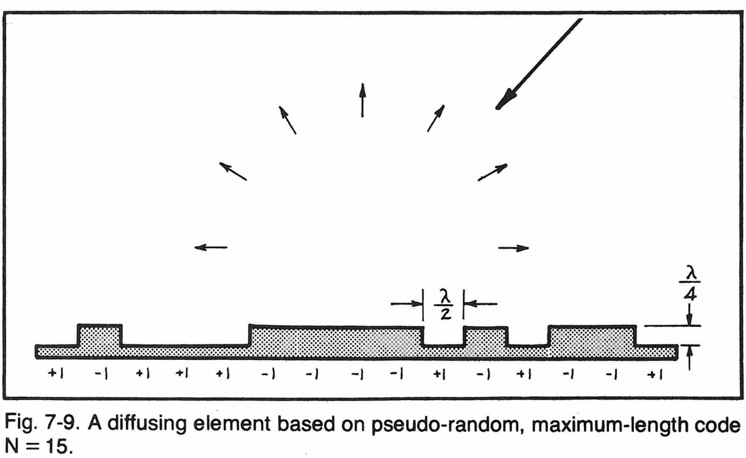

Fig. 7-9. A diffusing element based on pseudo-random, maximum-length code N = 15.

below the design frequency. Arrays based on different design frequencies would be necessary to cover an appreciable portion of the audible spectrum.

In the search for surface structures to give better sound diffusion over a greater frequency range, it was found that quadratic-residue sequences of elementary number theory gave even better results.[9] Diffusers of this type are being applied to concert hall ceilings to provide lateral reflections.

An excellent detailed discussion of the diffusion of sound in small rooms is given in reference 10.

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6. Gilford, Christopher. *Acoustics For Radio and Television Studios.* IEE Monograph Series II, Peter Perigrinus, Ltd., London (1972) pp.208.
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8. Schroeder, M.R. *Diffuse Sound Reflection By Maximum-Length Sequences.* Jour. Acous. Soc. Am., Vol. 57, No. 1 (January 1975) pp.149-150.
9. Schroeder, M.R. *Binaural Dissimilarity and Optimum Ceilings For Concert Halls.* Jour. Acous. Soc., Am., Vol. 64, No. 4 (April 1979) pp.958-963.
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8

Control of Interfering Noise

There are four basic approaches to reducing noise in a listening room or a recording studio:

(1) Locating the room in a quiet place,

(2) Reducing the noise energy within the room,

(3) Reducing the noise output of the offending source,

(4) Interposing an insulating barrier between the noise and the room.

Locating a listening room or a studio away from outside interfering sounds is a luxury few can enjoy because of the many factors (other than acoustical) involved in site selection. If it is a listening room and a part of a residence, due consideration must be given to serving the other needs of the family—at least if some degree of peace is to prevail. If the room in question is a recording or broadcast studio it is probably a part of a multipurpose complex and the noises originating from business machines, air conditioning equipment or foot traffic within the same building, or even sounds from other studios, may dominate the situation.

NOISE SOURCES, AND SOME SOLUTIONS

Protecting the room from street traffic noise is becoming more difficult all the time. It is useful to remember that doubling the distance from a noisy street or other sound source reduces the level of airborne noise approximately 6 dB. Shrubbery and trees may help

in shielding from street sounds; a cypress hedge 2 ft thick gives about a 4 dB reduction.

The level of noise which has invaded a room by one means or another can be reduced by introducing sound-absorbing material into the studio. For example, if a sound level meter registers a noise level of 45 dB inside a studio, this level might be reduced to 40 dB by covering the walls with great quantities of absorbing materials. Going far enough in this direction to reduce the noise significantly, however, would probably make the reverberation time too short. The control of reverberation must take priority. The amount of absorbent installed in the control of reverberation will reduce the noise level only slightly and beyond this we must look to other methods for further noise reduction.

Reducing the noise output of the offending source, if accessible and if possible, is the most logical approach. Traffic noise on a nearby street or airplanes overhead may be beyond control, but the noise output of a ventilating fan might be reduced 20 dB by the installation of a pliant mounting or the separation of a metal air duct with a simple canvas collar. Installing a carpet in a hall might solve a foot traffic noise problem or a felt pad a typewriter noise problem. In most cases working on the offending source and thus reducing its noise output is far more productive of results than corrective measures at or within the room in question.

As for terminology, we can say that a wall, for example, must offer a given "transmission loss" to sound transmitted through it as shown in Fig. 8-1. An outside noise level of 80 dB would be reduced to 35 dB by a wall having a transmission loss of 45 dB. A 60 dB wall would reduce the same noise level to 20 dB if no "flanking" or

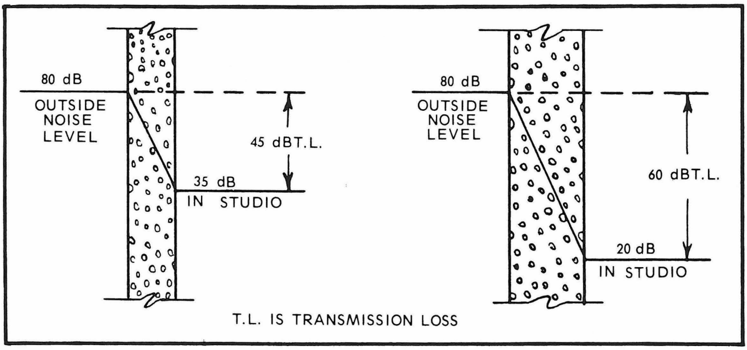


Fig. 8-1. The difference between the outside noise level and the desired noise level inside determines the required transmission loss of the wall.

bypassing of the wall by other paths is present. We say that the wall "attenuates" the sound or that it "insulates" the interior from the outside noise. The walls, floor, and ceiling of the listening room or studio must give the required transmission loss to outside noises, reducing them to tolerable levels inside the room. Some of the principles involved in solving this problem will now be considered.

Noise may invade a studio or other room in the following ways:

(a) Airborne,
(b) Transmitted by diaphragm action of large surfaces, or
(c) Transmitted through solid structures, or
(d) Combination of all three.

AIRBORNE NOISE

A heavy metal plate with holes to the extent of 13 percent of the total area may transmit as much as 97 percent of the sound impinging on it. The amount of sound which can pass through a small crack or aperture in an otherwise solid wall is astounding. A crack under a door or loosely fitting electrical service box can compromise the insulating properties of an otherwise excellent structure. Airtightness is especially necessary to insulate against airborne noises.

NOISE CARRIED BY STRUCTURE

Unwanted sounds can invade an enclosure by mechanical transmission through solid structural members of wood, steel, or masonry. Air conditioner noises can be transmitted to a room by the air in the ducts, by the metal of the ducts themselves, or both. We are all familiar with the excellent sound-carrying capabilities of water pipes and plumbing fixtures.

It is very difficult to make a solid structure vibrate by airborne noise falling upon it because of the inefficient transfer of energy from tenuous air to a dense solid. On the other hand, a motor bolted to a floor, a slammed door or an office machine on a table with legs on the bare floor can cause the structure to vibrate very significantly. These vibrations can travel great distances through solid structure with little loss. In fact, with wood, concrete, or brick beams, longitudinal vibrations are attenuated only about 2 dB in 100 ft. Sound travels in steel about 20 times as far for the same loss! Although joints and cross-bracing members increase the transmission loss, it is still very low in common structural configurations.

NOISE TRANSMITTED BY DIAPHRAGM ACTION

Although very little airborne sound energy is transmitted directly to a rigid structure, airborne sound can set a wall to vibrating as a diaphragm and the wall, in turn, can transmit the sound through the interconnected solid structure. Such structure-borne sound may then cause another wall at some distance to vibrate, radiating noise into the room we are interested in protecting. Thus two walls interconnected by solid structure may serve as a coupling agent between exterior airborne noise and the interior of the listening room or studio itself.

SOUND-INSULATING WALLS

For insulating against outside airborne sounds, the general rule is the heavier the wall the better. The more massive the wall, the more difficult it is for sound waves in air to move it to and fro. Figure 8-2 shows how the transmission loss of a rigid, solid wall is related to the density of the wall. The wall weight in Fig. 8-2 is expressed as so many pounds per square foot of surface, sometimes called the "surface density." For example, if a 10 × 10 ft concrete block wall weighs 2000 lb, the "wall weight" would be 2000 lb per 100 sq ft, or 20 lb per sq ft. The thickness of the wall is not directly considered.

From Fig. 8-2 we can see that the higher the frequency, the greater the transmission loss, or in other words, the better the wall is as a barrier to outside noises. The line for 500 Hz is made heavier than the lines for other frequencies as it is common to use this

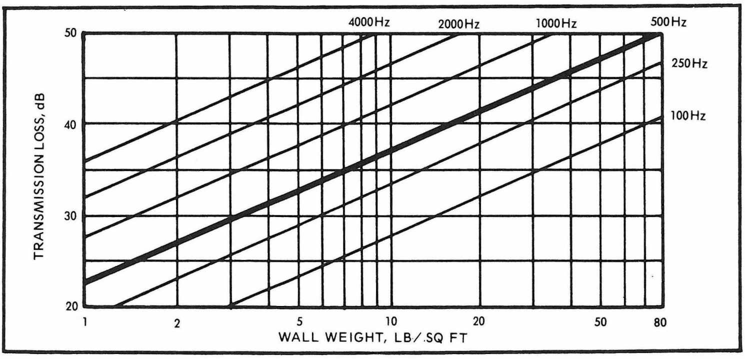


Fig. 8-2. The **mass** of the material in a barrier rather than the **kind** of material determines the transmission loss of sound going through the barrier. The transmission loss is also dependent on frequency although values at 500 Hz are commonly used in casual estimates. The wall weight is expressed in pounds per square foot of wall surface.

frequency for casual comparisons of walls of different materials. However, one should never forget that below 500 Hz the wall is less effective and for frequencies greater than 500 Hz it is more effective as a sound barrier.

The transmission losses indicated in Fig. 8-2 are based on the mass of the material rather than the kind of material. The transmission loss through a layer of lead of certain thickness can be matched by a plywood layer about 95 times thicker. But note that doubling the thickness of, say, a concrete wall, would increase the transmission loss only about 5 dB.

A discontinuous structure such as bricks set in lime mortar conducts sound less efficiently than a more homogenous material like concrete or steel. Unbridged air cavities between walls are very effective, but completely unbridged cavities are unattainable and only in the case of two separate structures, each on its own foundation, is this unbridged condition approached.

Porous Materials

Porous materials such as fiberglass (rockwool, mineral fiber) are excellent sound absorbers and good heat insulators but of limited value in insulating against sound. Using fiberglass to reduce sound transmission will help to a certain extent, but only moderately. The transmission loss for porous materials is directly proportional to the thickness traversed by the sound. This loss is about 1 dB (100 Hz) to 4 dB (3000 Hz) per inch of thickness for a dense, porous material (rockwool, density 5 lb/cu ft) and less for lighter material. Note that this direct dependence of transmission loss on thickness for porous materials is in contrast to the transmission loss for solid, rigid walls, which is approximately 5 dB for each doubling of the thickness.

Sound Transmission Classification (STC)

The solid line of Fig. 8-3 is simply a replotting of data from the mass law graphs of Fig. 8-2 for a wall weight of 10 lb per sq ft. If the mass law were perfectly followed, we would expect the transmission loss of a practical wall of this density to vary with frequency as shown by the solid line. Unfortunately, things are seldom this simple and we will find that actual measurements of transmission loss on this wall might be more like the broken line of Fig. 8-3. These deviations reflect resonance and other effects in the wall panel which are not included in the simple mass law concept.

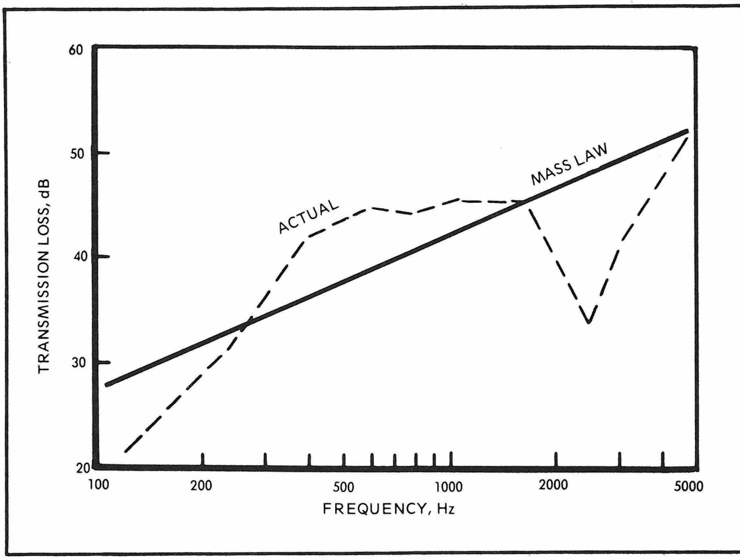


Fig. 8-3. Actual measurements of transmission loss in walls often deviate considerably from the mass law (Fig. 8-2) because of resonances and other effects.

Because of such commonly occurring irregularities, it would be of great practical value to agree on some arbitrary procedure of arriving at a single number which would give a reasonably accurate indication of the sound transmission loss characteristics of a wall. This has been done in a procedure specified by the American Society For Testing Materials in which the measured graph of a wall would be placed in a certain Sound Transmission Class (STC) by comparison to a reference graph (STC contour). The details of this procedure are beyond the scope of this book but the results of such classification have been applied to walls of various types to be described for ready comparison. An STC rating of 50 dB for a wall would mean that it is better in insulating against sound than a wall of STC 40 dB. It is not proper to call STC ratings "averages" but the whole procedure is to escape the pitfalls of averaging dB transmission losses at various frequencies.

COMPARISON OF WALL STRUCTURES

Figure 8-4 gives the measured performance of a 4″ concrete block wall as a sound barrier. It is interesting to note that plastering both sides increases the transmission loss of the wall from STC 40 dB to 48 dB. Figure 8-5 shows a considerable improvement in doubling the thickness of the concrete block wall. In this case the

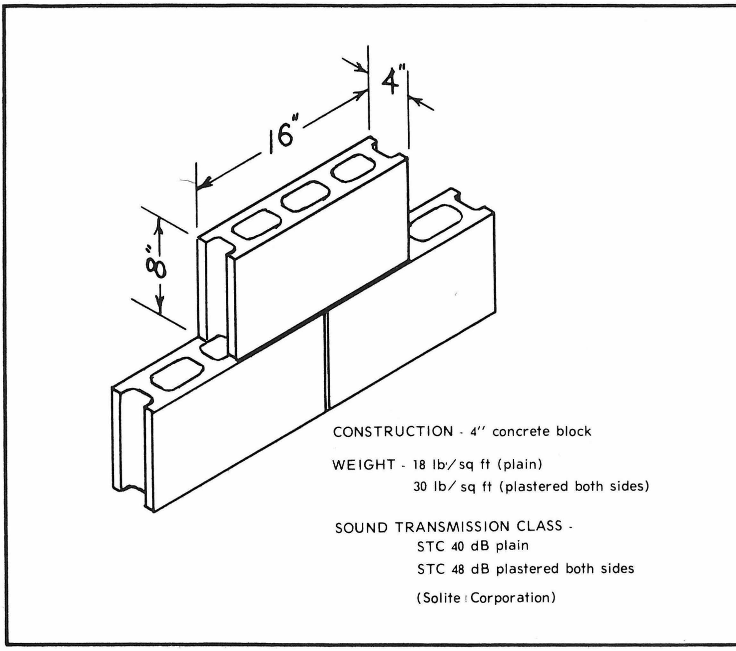


Fig. 8-4. 4" concrete block.

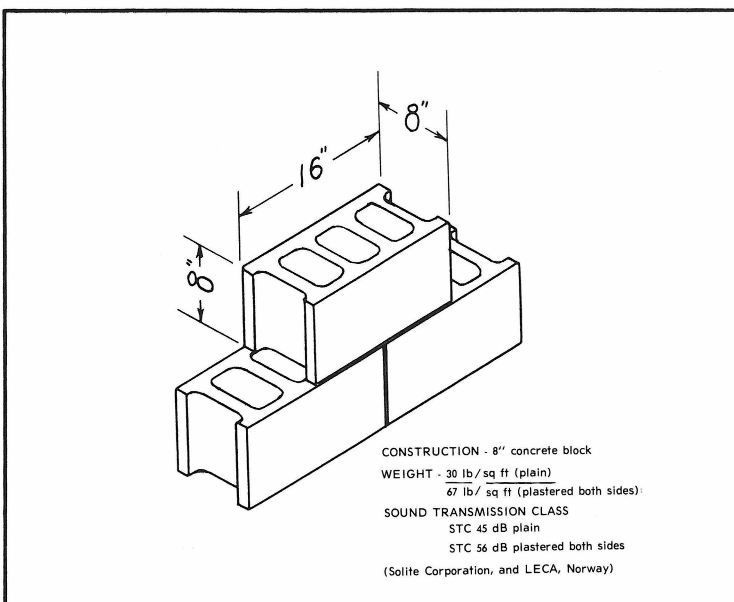
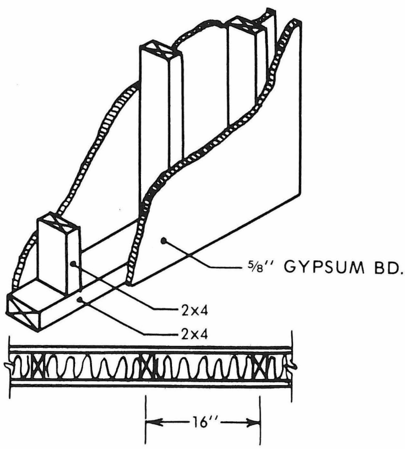


Fig. 8-5. 8" concrete block.



CONSTRUCTION - Standard stud partition

WEIGHT - 7.3 lb/sq ft

SOUND TRANSMISSION CLASS

STC 34 dB without fiberglass

STC 36 dB with 3½" fiberglass

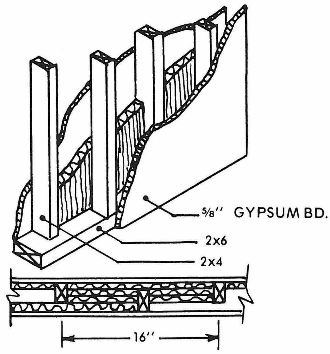
(Owens-Corning Fiberglas Corporation)

Fig. 8-6. Standard stud partition.

STC 45 dB is improved 11 dB by plastering both sides. In Fig. 8-6 we have the very common 2 × 4 frame construction with ⅝" gypsum board covering. The STC 34 dB without fiberglass between is improved only 2 dB by filling the cavity with fiberglass material, a meager improvement which would probably not justify the added cost.

Figure 8-7 describes a very useful and inexpensive type of wall of staggered stud construction. Here the inherently low coupling between the two independent wall diaphragms is further reduced by filling the space with fiberglass building material. Attaining the full STC 52 dB rating would require careful construction to insure that the two wall surfaces are truly independent and not "shorted out" by electrical conduits, outlet boxes, etc.

The last wall structure to be described is the double wall construction of Fig. 8-8. The two walls are entirely separate, each having its own 2 × 4 plate. Without fiberglass this wall is only 1 dB better than the staggered stud wall of Fig. 8-7 but by filling the inner space with building insulation, STC ratings up to 58 dB are possible.



CONSTRUCTION - Staggered stud partition

WEIGHT - 7.2 lb/ sq ft

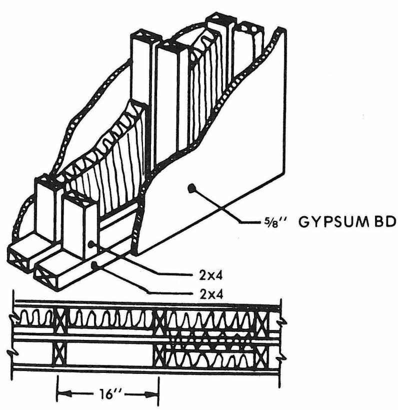
SOUND TRANSMISSION CLASS

STC 42 dB without fiberglass

STC 46 to 52 dB with fiberglass

(Owens-Corning Fiberglas Corporation)

Fig. 8-7. Staggered stud partition.



CONSTRUCTION - Double wall

WEIGHT - 7.1 lb/ sq ft

SOUND TRANSMISSION CLASS

STC 43 dB without fiberglass

STC 55 dB with 3½" fiberglass

STC 58 dB with 9" fiberglass

(Owens-Corning Fiberglas Corporation)

Fig. 8-8. Double wall.

Earlier in this chapter it was stated that porous sound absorbing materials are of limited value in insulating against sound. This is true when normal transmission loss is considered but in structures as those in Figs. 8-7 and 8-8, such porous materials have a new contribution to make in absorbing sound energy in the cavity. This can improve the transmission loss in some wall structures by as much as 15 dB, principally by reducing resonances in the space between the walls, while in others the effect is negligible. The low density mineral fiber batts commonly used in building construction are as effective as the high density boards and are much cheaper. Mineral fiber batts within a wall may also meet certain fire-blocking requirements in building codes.

The staggered stud wall and the double wall, on the basis of mass alone, would yield a transmission loss of only about 35 dB (Fig. 8-2). The isolation of the inner and the outer walls from each other and the use of insulation within have increased the wall effectiveness by 10 or 15 dB.

DOUBLE WINDOWS

Between control room and studio a window is quite necessary and its sound transmission loss should be comparable to that of the wall itself. A well-built staggered stud or double wall may have an STC of 50 dB. To approach this performance with a window requires very careful design and installation.

A double window is most certainly indicated; a triple window adds little more. The mounting must minimize coupling from one wall to the other. One source of coupling is the window frame, another is the stiffness of the air between the glass panels. The plan of Fig. 8-9 is a practical solution to the double-window problem for concrete block walls. Figure 8-9B is an adaptation to the staggered stud construction. In the latter there are, in effect, two entirely separate frames, one fixed to the inner and the other to the outer staggered stud walls. A felt strip may be inserted between them to insure against accidental contact.

Heavy plate glass should be used, the heavier the better. There is a slight advantage in having two panes of different thickness. If desired, one glass can be inclined to the other to control light or external sound reflections but this will have negligible effect on the transmission loss of the window itself. The glass should be isolated from the frame by rubber or other pliable strips. The spacing between the two glass panels has its effect, that is, the greater the spacing the greater the loss; but there is little gain in

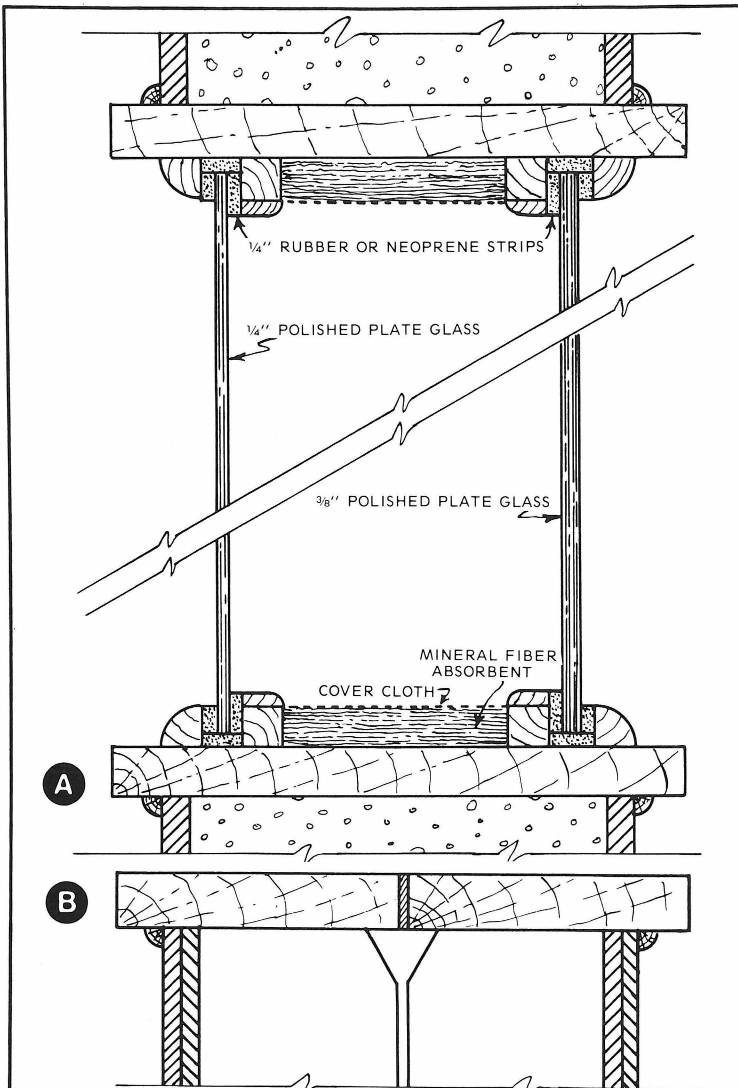


Fig. 8-9. Plans for double glass windows (A) for concrete block wall and (B) for staggered stud frame wall. Important features include the resilient mounting of the plate glass and the sound absorbing material around the periphery of the space between the two glass sheets.

going beyond 8 inches nor serious loss in dropping down to 4 or 5 inches.

The absorbent material between the planes in the design of Fig. 8-9 discourages resonances in the air space. This adds signifi-

cantly to the overall insulation efficiency of the double window and it should extend completely around the periphery of the window. If the double window of Fig. 8-9 is carefully constructed, sound insulation should approach that of an STC 50 dB wall but will probably not quite reach it. For the staggered stud wall in which a double window is to be placed the use of 2 × 8 instead of the 2 × 6 plate will simplify mounting of the inner and outer window frames.

Prefabricated double glass windows are available commercially, one of which is rated at STC 49 dB.

SOUND-INSULATING DOORS

The transmission loss of a door is determined by its mass, stiffness, and air-tightness. An ordinary household panel door hung in the usual way might offer less than 20 dB sound insulation. Increasing the weight and taking reasonable precautions on seals might gain another 10 dB, but a door to match a 50 dB wall requires great care in design, construction, and maintenance. Steel doors or patented acoustical doors giving specified values of transmission loss are available commercially but they are quite expensive. To avoid the expense of doors having high transmission loss, "sound

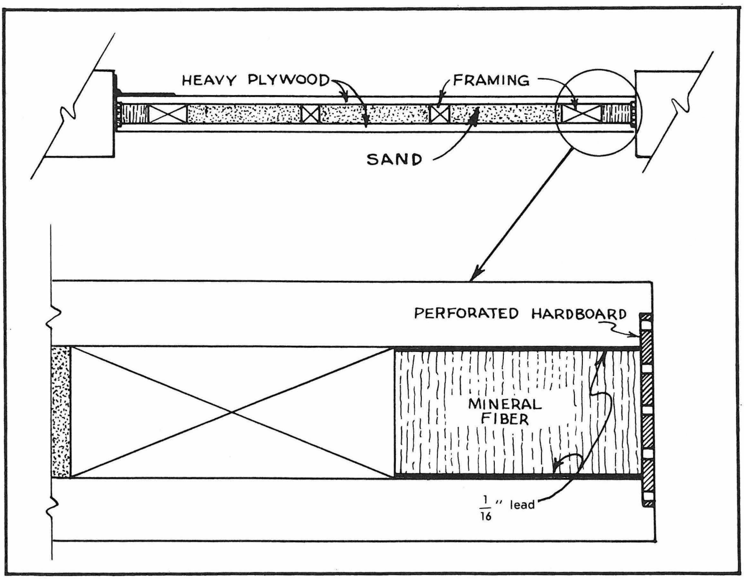


Fig. 8-10. A reasonably effective and inexpensive "acoustic" door. Dry sand between the plywood faces adds to the mass and thus the transmission loss. Sound traveling between the door and jamb tends to be absorbed by the absorbent door edge.

locks" are commonly used. These small vestibules with two doors of medium transmission loss are very effective and convenient.

Doors having good insulating properties can be constructed if the requirements of mass, stiffness, and airtightness are met. Figure 8-10 suggests one inexpensive approach to the mass requirement, filling a hollow door with sand. Heavy plywood (¾ inch) is used for the door panels.

Achieving a good seal around a "soundproof" door can be very difficult. Great force is necessary to seal a heavy door. Wear and tear on pliant sealing strips can destroy their effectiveness, especially at the floor where foot-wear is a problem. The detail of Fig. 8-10 shows one approach to the sound leakage problem in which a very absorbent edge built around the periphery of the door serves as a trap for sound traversing the crack between door and jamb. This absorbent trap could also be imbedded in the door jamb. Such a soft trap could also be used in conjunction with one of the several types of seals.

Figure 8-11 shows a do-it-yourself door seal which has proved reasonably satisfactory. The heart of this seal is a rubber or plastic tubing an inch or less in outside diameter with a wall thickness of about 3/32 inch. The wooden nailing strips hold the tubing to the door frame by means of a canvas wrapper. A raised sill is required at the floor if the tubing method is to be used all around the door (or another type of seal such as weatherstripping could be utilized at the bottom of the door). An advantage of tubing seal is that the degree of compression of the tubing upon which the sealing properties depend is available for inspection.

A complete door plan patterned after BBC practice is shown in Fig. 8-12. It is based upon a 2" thick solid slab door and utilizes a magnetic seal such as used on refrigerator doors. The magnetic material is barium ferrite in a PVC (polyvinylchloride) rod. In

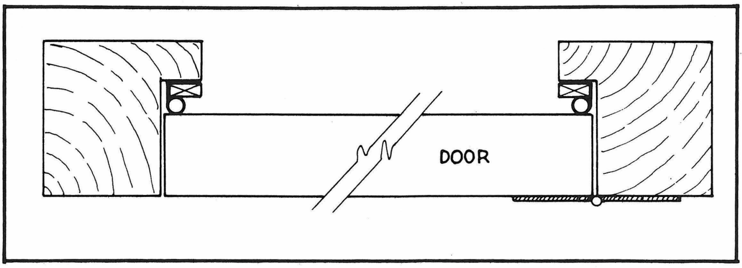


Fig. 8-11. A door may be sealed by compressible rubber or plastic tubing held in place by a fabric wrapper.

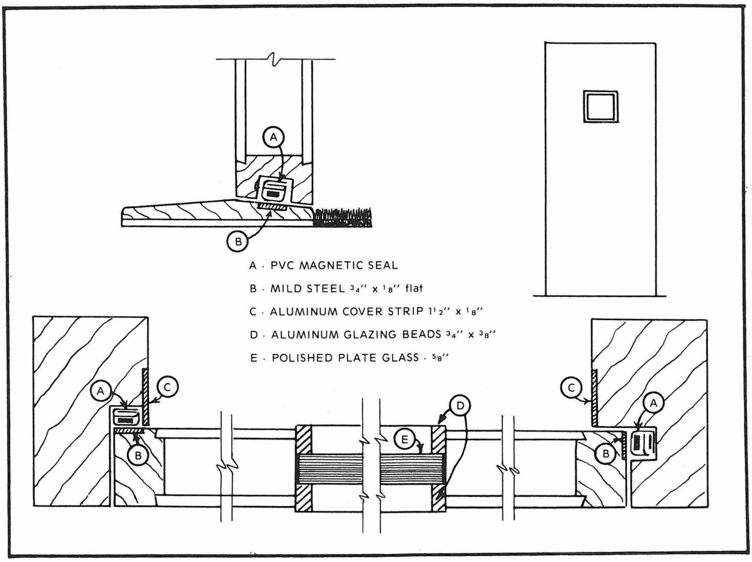


Fig. 8-12. A BBC door design utilizing magnetic seals of the type used on refrigerator doors.

pulling toward the mild steel strip, a good seal is achieved. The aluminum strip "C" decreases sound leakage around the periphery of the door.

It is possible to obtain a very slight acoustical improvement and, to some at least, an improvement in appearance by padding both sides of a door. A plastic fabric over 1" foam rubber sheet may be "quilted" with upholstery tacks. (Refer to Figs. 16-1 through 16-5 for an example.)

NOISE AND ROOM RESONANCES

Room resonances can affect the problem of outside noise in a studio. Any prominent modes persisting in spite of acoustic treatment make a room very susceptible to interfering noises having appreciable energy at these frequencies. In such a case a feeble interfering sound could be augmented by the resonance effect to a very disturbing level.

9

Acoustical Materials and Structures

The first problem the do-it-yourself worker encounters in contemplating the acoustical treatment of an audio room is what materials to use and where to get them. One way of gathering such information is to visit your friendly neighborhood building materials supplier and ask questions. Often when "acoustical materials" are mentioned, enthusiastically delivered information on acoustical tile and ceiling boards is readily elicited. Beyond that the information received usually varies from vague to opaque. Another approach is to write to manufacturers, but this first requires identifying and locating the manufacturers. The recommended approach is to obtain a copy of the latest edition of the *Compendium of Materials For Noise Control.*[1] This 380 page, 8½ × 11 inch government publication summarizes basic acoustical data on materials offered by 146 different companies. Not only that, the addresses of all 146 companies are included as well as a useful section on acoustical fundamentals running to almost 100 pages.

ABSORPTION COEFFICIENT

The absorption coefficient of an acoustical material is defined as the ratio of sound absorbing effectiveness, at a specific frequency, of a unit area of acoustical absorbent to a unit area of a perfectly absorptive material. This absorption coefficient varies both with frequency and the angle of incidence. Measurements on a given material are made at standard frequencies of 125, 250, 500,

1000, 2000, and 4000 Hz. The angle of incidence problem is solved by making measurements in a reverberation chamber which assures essentially random incidence. The general procedure is to measure the rate of sound decay in the reverberation chamber both with and without a standard 72 sq.ft. sample of the material under test on the concrete floor.

Young has pointed out a long-standing and widespread confusion in the field of acoustics concerning the sound absorption coefficient.[2] There really are two kinds, one based on the arithmetic mean reflection coefficient of the several sound absorbing surfaces, a, and the other the geometric mean reflection coefficient, α, which are related by:

$$a = -\log_e (1-\alpha)$$

in which

a = Sabine absorption coefficient

α = energy absorption coefficient

We can skirt this problem by concentrating our attention on the Sabine coefficient, a, which is actually what is measured and published in various tables.

STANDARD MOUNTINGS

It is common knowledge that the way a given sound absorbing material is mounted greatly affects its efficiency as an absorber. Tables of sound absorption coefficients will usually specify the mounting of the material that applies. The Compendium lists eight standard mountings, but only two of them will cover most practical applications. These two, Mounting 4 and Mounting 7 are illustrated in Fig. 9-1. The material being measured is layed directly on the concrete floor of the reverberation room to conform to Mounting 4. An acoustical tile, for example, cemented to a wall can be taken as Mounting 4. Automobile buffs can remember this by associating it with "four on the floor". Mounting 7 covers suspended or lay-in acoustical ceilings with a nominal depth of 16 inches. Strictly speaking, the coefficients published for a given material apply only to this airspace. If another mounting depth is used, measured coefficients are not available, which is one of the problems of estimating absorbence of practical spaces.

In listings of sound absorbing materials the noise reduction coefficient (NRC) is usually given as a single number rating for the material. Sometimes a figure such as NRC = 0.90 occurs alone and at other times is added to the listing of absorption coefficients at the

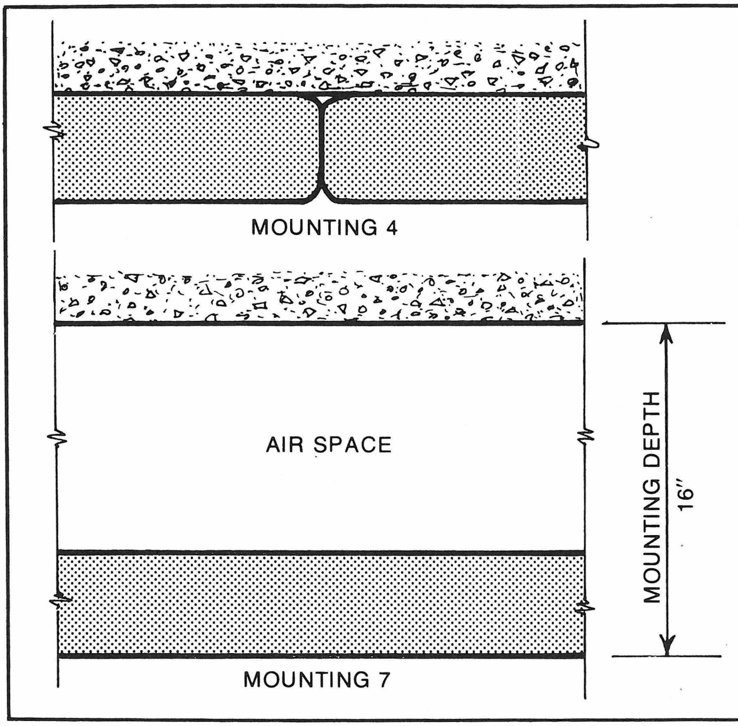


Fig. 9-1. Commonly used standard mountings associated with listings of absorption coefficients. With Mounting 4 the material is flat against the backing. Mounting 1 (not shown) is similar to Mounting 4 but with a 1/16" air space resulting when acoustical materials are cemented to a surface. Mountings 1 and 4 are essentially the same. Mounting 7 applies to suspended ceilings with lay-in panels.

six standard frequencies. The NRC is simply the average of the coefficients for 250, 500, 1000, and 2000 Hz, dropping off the outside frequencies of 125 and 4000 Hz. It is rounded off to the nearest 0.05. Because of the crude method used to obtain the NRC rating, it is of limited value in treating audio rooms but helpful, at times, in screening long lists of possible materials.

POROUS ABSORBERS

Sound is absorbed by the tiny interstices (small cracks or spaces) of porous materials such as felt, carpets, drapes, fibrous batts, and boards made of cellulose or mineral fibers. In Fig. 1-17 we saw what happens to the sound energy striking a material. Some of it is reflected, that which penetrates is partly absorbed and part of it comes out the other side. The absorption efficiency of materials

depending upon the trapping and dissipating of sound energy in tiny pores can be seriously impaired if the surface pores are filled so that penetration is limited. For example, coarse concrete block has many such pores and is a fair absorber of sound. Painting that block fills the surface pores and greatly reduces sound penetration, and thus absorption. However, if spray painted with a non-bridging paint the absorption may be reduced very modestly. Acoustical tile painted at the factory minimizes the problem of reduced absorption. Under certain conditions a painted surface can reduce porosity but act as a diaphragm which may actually become a fair absorber on a different principle, that of a damped vibrating diaphragm.

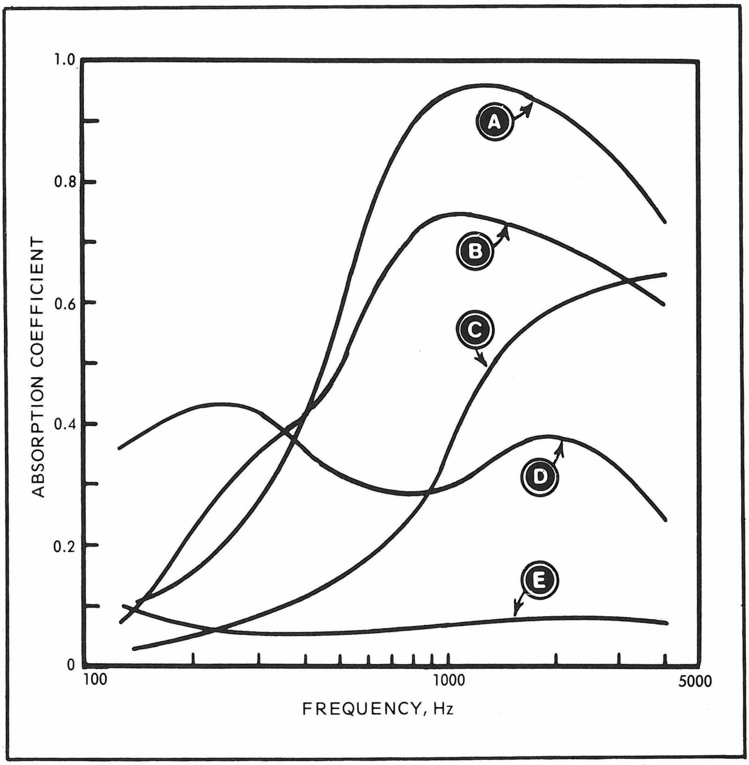


Fig. 9-2. Sound absorption coefficients of typical porous materials, A, B, and C, show a similarity in general shape. Good high frequency absorption and poor low frequency absorption characterize porous absorbers.

(A) High grade acoustical tile.
(B) Medium weight (14 oz/sq yd) velour draped to half area.
(C) Heavy carpet on concrete without padding.
(D) Coarse concrete blocks, unpainted.
(E) Coarse concrete blocks, painted.

(Data from the *Compendium*[1])

In the first radio broadcasting studios the acoustical treatment was an overuse of carpeted floors and drapes which emphasized a serious shortcoming of most porous absorbers, that of poor low frequency absorption. Tiles of cellulose fiber with perforated faces became the next style of treatment, but they were also deficient in low frequency absorption. Too enthusiastic use of porous absorbers, not only during the early days but even today, causes overabsorption of the high frequency sound energy, without touching the major problem of room acoustics, low frequency standing waves.

To show the general similarity of the absorption characteristics of sound absorbers depending on porosity for their effectiveness, a side-by-side comparison is made in Fig. 9-2. The acoustical tile, drapes, and carpet show highest coefficients above 500 Hz and relatively low coefficients in the low frequency region dominated by room modes. Course concrete blocks show the usual high frequency peak, but also an unusual peak around 200 Hz, the mechanism of which seems most un-porouslike.

ACOUSTICAL TILE

During the 1960s and 1970s many top line manufacturers of acoustical materials offered their competitive lines of 12″×12″ acoustical tiles. Surface treatments of the tiles included even-spaced holes, random holes, slots, or fissured or other special textures. Today we observe that the Compendium lists no 12″ × 12″ tiles for Mounting 4. In spite of this they continue to be available from local building material suppliers. Such tiles are reputable products for noise and reverberation control as long as they are used with full knowledge of their limitations which are roughly described by curve A of Fig. 9-2. One of the problems of using acoustical tile in critical situations is that absorption coefficients are rarely available for the specific tile obtainable. Going back into the earlier literature, the average of the coefficients for eight cellulose and mineral fiber tiles of ¾″ thickness is shown in Fig. 9-3. The range of the coefficients is indicated by the vertical lines. The average points could be used for ¾″ tile for which no coefficients are available. Coefficients 20% lower would be a fair estimate for ½″ tiles.

CARPET

Carpet dominates the acoustical situation when used in an audio room. Its unbalanced characteristic often provides most of the absorption needed in the high frequency region leaving little left to apply to walls to prevent flutter echoes and to aid diffusion. In this

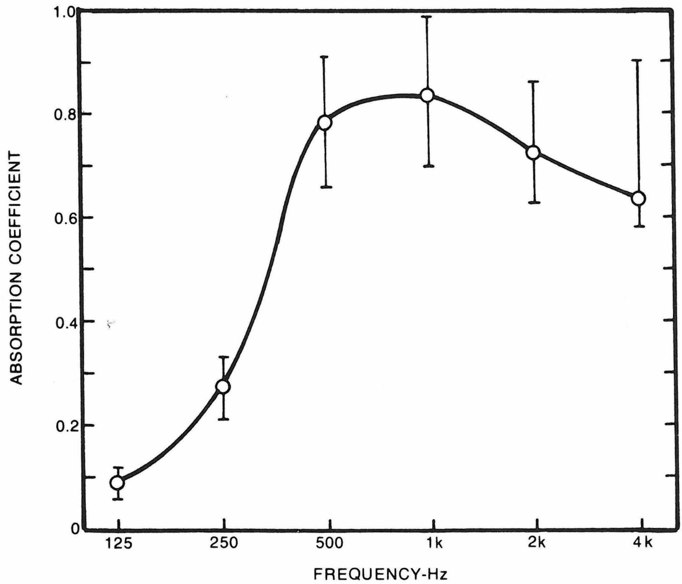


Fig. 9-3. The average absorption characteristics of 8 acoustical tile brands of ¾″ thickness. The vertical lines show the spread of the data.

way, overabsorption at high frequencies and underabsorption at low frequencies go hand in hand when carpet is used, especially for full wall-to-wall coverage.

There are many different types of carpet available and coefficients to go with a specific carpet are usually not available. We are left the task of placing a selected carpet in a broad category for which coefficients are available. A latex backing has the effect of substantially increasing absorption above 500 Hz and decreasing it modestly below 500 Hz as compared to the same carpet without latex backing. Carpet underlayment has a significant effect, whether foam rubber, sponge rubber, felt, or polyurethane. Open cell rubber and foam provides better absorption than the closed cell type.

Figure 9-4 illustrates the range of absorption coefficients for three carpet/underlay conditions. Further information on the effects of carpet pile and underlayment may be found in the article listed in reference 3. When all is said and done, however, a large element of judgment is involved in deciding upon coefficients to use for a selected carpet.

DRAPERIES

The absorption of drapery materials is a function of (a) type and weight of the material, (b) the degree of fold, and (c) the distance from the wall.[5]. The heavier the material the greater the sound absorption. Drapes are porous if air can be blown through the material and it is the porosity which allows penetration as well as absorption of sound. A high degree of fullness of the fold makes the material absorb sound as though it were much thicker. An extreme case would bring the folds tightly together creating an effective thickness equal to the thickness of the fold. The effect of fold on one cotton drapery material is shown in Fig. 9-5. The absorption coefficient of this 14.7 ounce cotton cloth at 1 kHz is 0.27 for ⅞ drape but is 3 times higher or 0.81 for ½ drape, an increase of 200%. Just remember that it takes a lot more material folded deeply to present a given area to the sound than a drape with no fold.

The distance a drape is positioned from the wall is a major influence on its absorptivity. As sound is reflected from a hard wall, the pressure is highest at the wall and air particle velocity is zero.

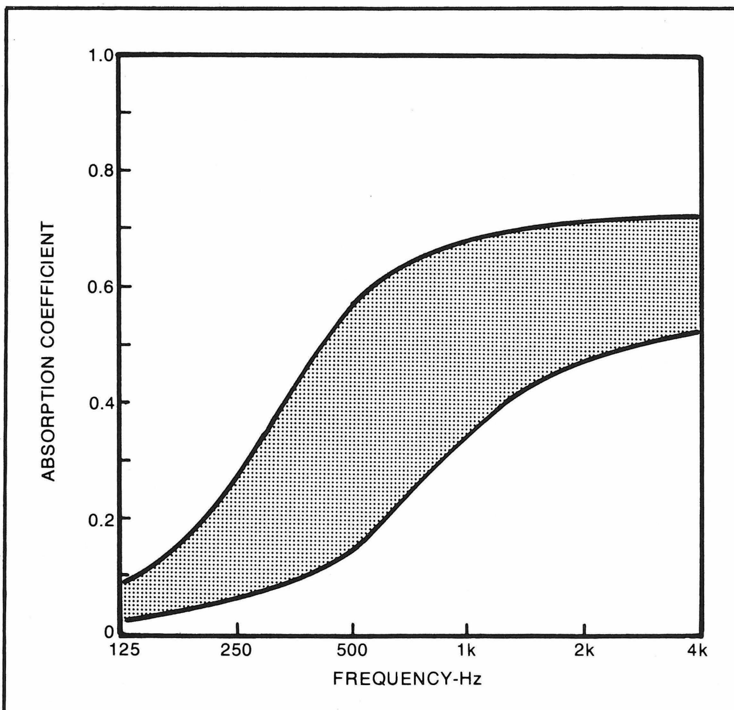


Fig. 9-4. Range of absorption coefficients for carpet listed in the *Compendium*.[1]

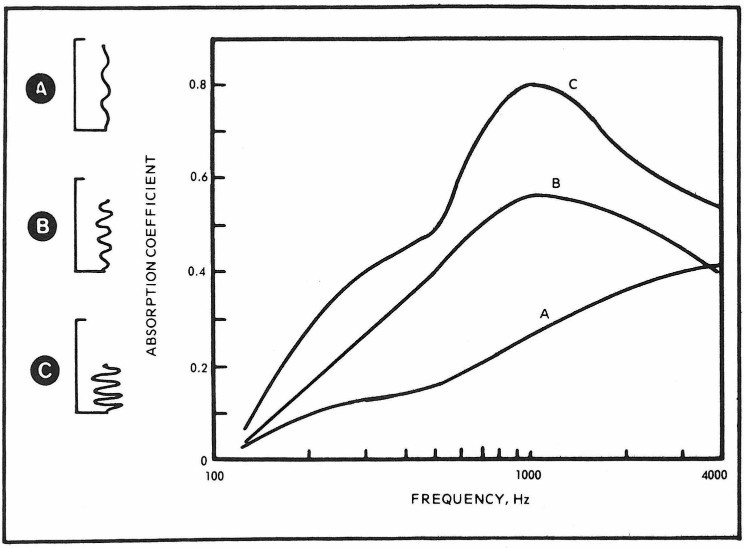


Fig. 9-5. The more drapery material is folded, the higher the absorption coefficient. Material: cotton cloth, 14.7 oz/sq yd.

(A) Draped to 7/8 area (almost flat)
(B) Draped to 3/4 area
(C) Draped to 1/2 area

(After Mankovsky[4]).

At one quarter wavelength ($\frac{\lambda}{4}$) from the wall the pressure is zero but the particle velocity is maximum. If the drapery material hangs $\frac{\lambda}{4}$ from the wall the frantic movement of air particles takes place in the fabric of the drape and greatest frictional losses occur. Consequently the absorption of sound will be greatest at the particular frequency that makes the spacing a quarter wavelength. The same thing happens at odd multiples of $\frac{\lambda}{4}$ such as $\frac{3\lambda}{4}, \frac{5\lambda}{4}, \frac{7\lambda}{4}$, etc.

It is easy to evaluate these peak absorption frequencies at $\frac{\lambda}{4}$ and odd multiples of $\frac{\lambda}{4}$ from Fig. 9-6. Let us consider one drape hung 18 inches from a flat, reflective surface and another at 6 inches from the wall. A horizontal line drawn at 18 inches and at 6 inches in Fig. 9-6 will identify the frequencies of maximum absorption. These frequencies are shown in Table 9-1.

A sketch of the theoretical absorption coefficient for draperies hung 18 inches and 6 inches from a hard, reflective wall is shown in Fig. 9-7. The maxima are assumed to reach a coefficient of 1.0 and the minima are assumed to go to zero as we consider only the wall

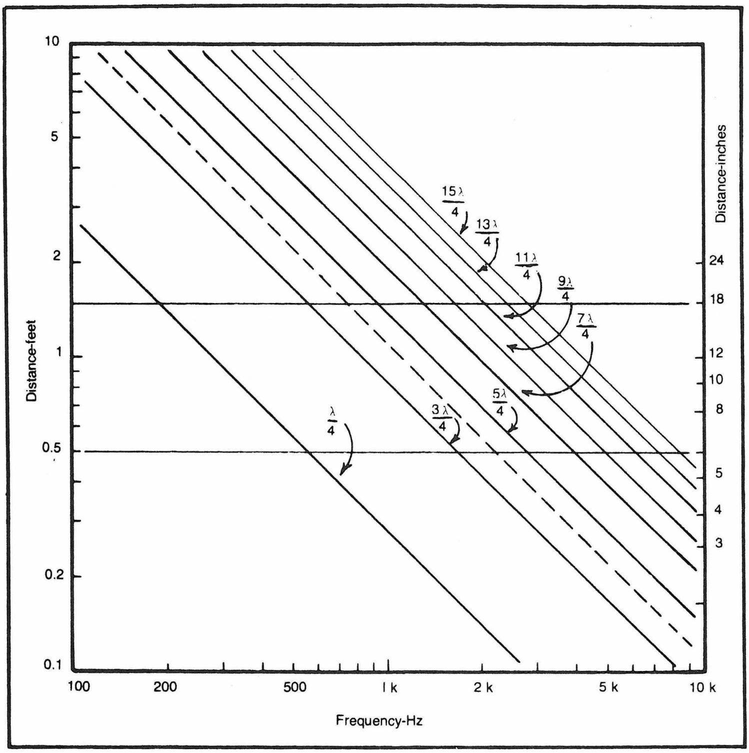


Fig. 9-6. A graph to simplify finding the frequency of the quarter wavelength and odd multiples of the quarter wavelength of sound as applied to the spacing of drapes from walls. The drapery absorption peaks occur at the odd quarter wavelength points, the minima at half wavelength points. The horizontal lines at 6″ and 18″ spacing are for the example of Table 9-1 and Fig. 9-7.

reflection effect. Increasing the spacing from the wall increases low frequency absorption.

The cyclic variation of absorption with frequency may cause a bit of preliminary anguish. Many acoustical measurements are made in octave bands. Tables of absorption coefficients are built on octave intervals. In Fig. 9-7 the ranges of the 6 most used octave bands are shown. Octave measurements made in a room having appreciable areas of draperies would tend to smear out the effect of the narrow minima. Further, drapes are usually used with some degree of folding and different parts of the fold are at different distances from the wall which further tends to smear the minima. We conclude that the typical theoretical variations in Fig. 9-7 seldom are delineated in practice and are rarely a source of difficulty.

Table 9-1. Frequencies of Maximum and Minimum Drapery Absorption.

Speed of sound=1130 ft/sec

	Odd	$\frac{\lambda}{4}$		$\frac{3\lambda}{4}$		$\frac{5\lambda}{4}$		$\frac{7\lambda}{4}$		$\frac{9\lambda}{4}$		$\frac{11\lambda}{4}$		$\frac{13\lambda}{4}$
	Even		$\frac{\lambda}{2}$		$\frac{2\lambda}{2}$		$\frac{3\lambda}{2}$		$\frac{4\lambda}{2}$		$\frac{5\lambda}{2}$		$\frac{6\lambda}{2}$	
Spacing=18″	Freq, Hz Max. Absorp.	188		565		942		1318		1695		2071		2448
Spacing=18″	Freq, Hz Min. Absorp.		377		753		1130		1507		1883		2260	
Spacing=6″	Freq, Hz Max. Absorp.	565		1695		2825		3955		5085		6215		7345
Spacing=6″	Freq, Hz Min. Absorp.		1130		2260		3390		4520		5650		6780	

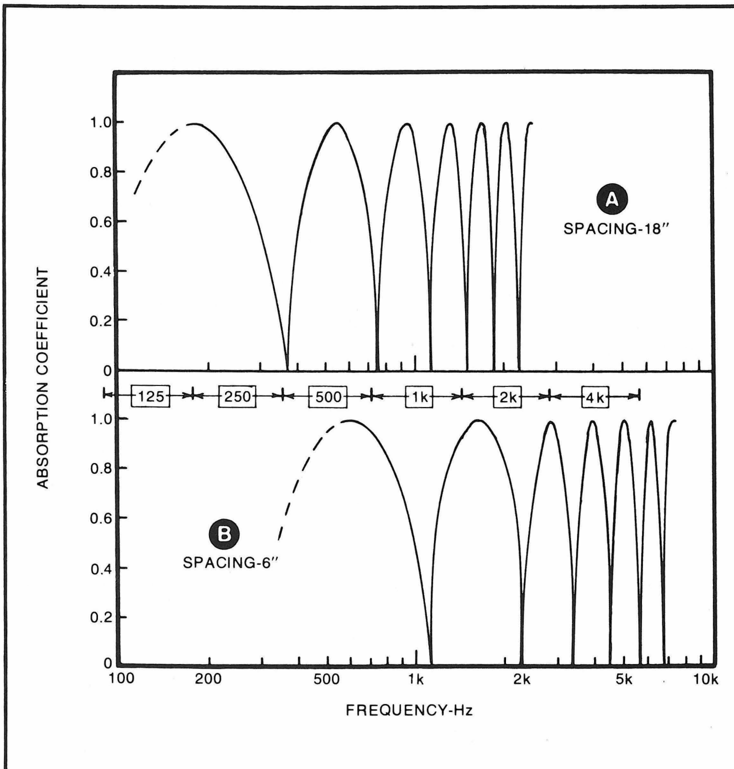


Fig. 9-7. Absorption peaks and minima for drapery material hung (A) 18″ from a reflecting wall, and (B) 6″ from reflective wall. Low frequency absorption is improved by greater spacings. The range of the six common octave measuring bands would tend to smear the precise, theoretical minima shown.

In Fig. 9-2, the B curve, taken from the *Compendium* (pg 55), is for medium velour, 14 oz/sq yd, draped to half area. What is not stated is how the velour is mounted. It is most likely that it was hung far away from walls and what was measured was the normal porosity type of absorption quite independent of the wall reflection effect. Therefore, by proper spacing of this velour from a reflecting wall, the low frequency deficiency could be filled in and the general level of absorption increased.

BASS TRAPS

While the concept of wall reflection, graphically portrayed in Fig. 9-6, is fresh in our minds, let us consider "bass traps" which act on similar principles. This phrase is applied to many kinds of low frequency sound absorbers, such as panel absorbers, but it should really be reserved for a special type of reactive cavity absorber which has been widely used. It has been described by Rettinger.[6] A true bass trap is shown in Fig. 9-8. It is simply a box or cavity of very critical depth but with a mouth opening of size to suit particular purposes. This is a tuned cavity of $\frac{\lambda}{4}$ depth at the design frequency at which maximum absorption is desired. Sound absorption at the lowest octave or two of the audible spectrum is often difficult to

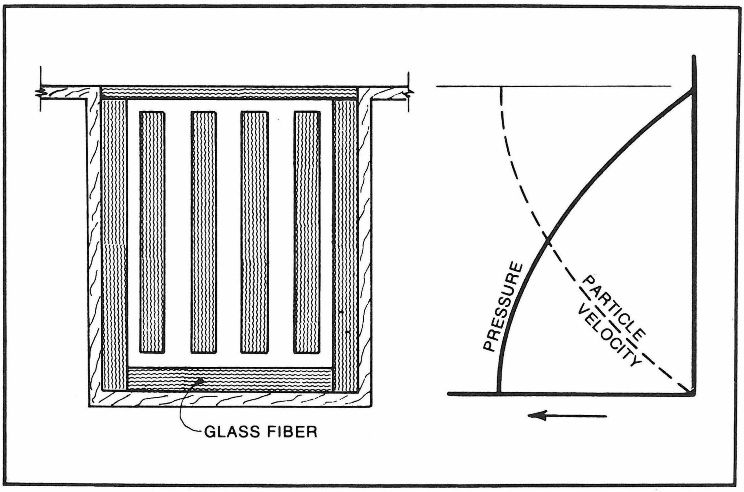


Fig. 9-8. The bass trap depends upon reflections of sound from the bottom for its action. The pressure for the frequency at which the depth is a quarter wavelength is maximum at the bottom and the particle velocity is zero at the bottom. At the mouth the pressure is zero (or very low) and the particle velocity is maximum. Absorbent placed where the particle velocity is maximum will absorb sound very effectively. The same action occurs at odd multiples of the quarter wavelength.

achieve. The bass trap is commonly used in recording studios to control standing waves at these bass frequencies.

The sound pressure at the bottom of the cavity is maximum at the $\frac{\lambda}{4}$ design frequency. The air particle velocity is zero at the bottom. At the mouth the pressure is zero and the particle velocity is maximum which results in two interesting phenomena. First, a glass fiber semi-rigid board across the opening offers great friction to the rapidly vibrating air particles resulting in maximum absorption at this frequency. In addition, the zero pressure at the opening constitutes a vacuum which tends to suck sound energy into this "sound sink" from surrounding areas. The bass trap's effect, then, is greater than its opening area would suggest because of this "vacuum cleaner" effect.

This bass trap effect, like the drape spaced from a reflective wall, occurs not only at $\frac{\lambda}{4}$ but at odd multiples of $\frac{\lambda}{4}$. For this reason, Fig. 9-6 is useful in determining where the maxima occur and Fig. 9-7 is instructive in giving a picture of bass trap absorption down through the audible spectrum. Figure 9-6 is carried down only to 100 Hz, however, and bass traps are commonly used a couple of octaves below this. Great trap depths are required for very low frequencies. For example, $\frac{\lambda}{4}$ for 40 Hz is 7 ft. Unused spaces above control room ceilings and between the inner walls and outer shells are often used for trap space.

GLASS FIBER: BUILDING INSULATION

Great quantities of glass fiber materials are used in the acoustical treatment of recording studios, control rooms, and public gathering spaces, some is special, high density material, some ordinary building insulation. In wood or steel stud single frame walls, double walls, and staggered stud walls thermal insulating batts are commonly used. This material usually has a density of about 1 lb/cu ft. Such material is often identified as R-11, R-19, or other such numbers. These R-prefix designations have to do with thermal insulating qualities, but are related to thickness. Thus R-8 is 2.5″ thick, R-11 is 3.5″, and R-19 is 6″ as listed in a very helpful Owens-Corning brochure.[7]

Building insulation installed within a wall increases its transmission loss a modest amount, primarily by reducing cavity resonance which would tend to couple the two wall faces at the resonance frequency of the cavity. A certain increase in the transmission

loss of the wall can also be attributed to attenuation of sound in passing through the glass fiber material, but this loss is small because of the low density of the material. Considering all mechanisms, the transmission loss of a staggered stud wall with a layer of gypsum board on each side can be increased about 7 dB by adding 3.5″ of building insulation. A double wall might show as much as a 12 dB increase by adding 3.5″ and 15 dB with 9″ of insulation. As far as wall transmission loss is concerned, using the denser, more expensive glass fiber between wall faces offers no appreciable advantage over ordinary building insulation.

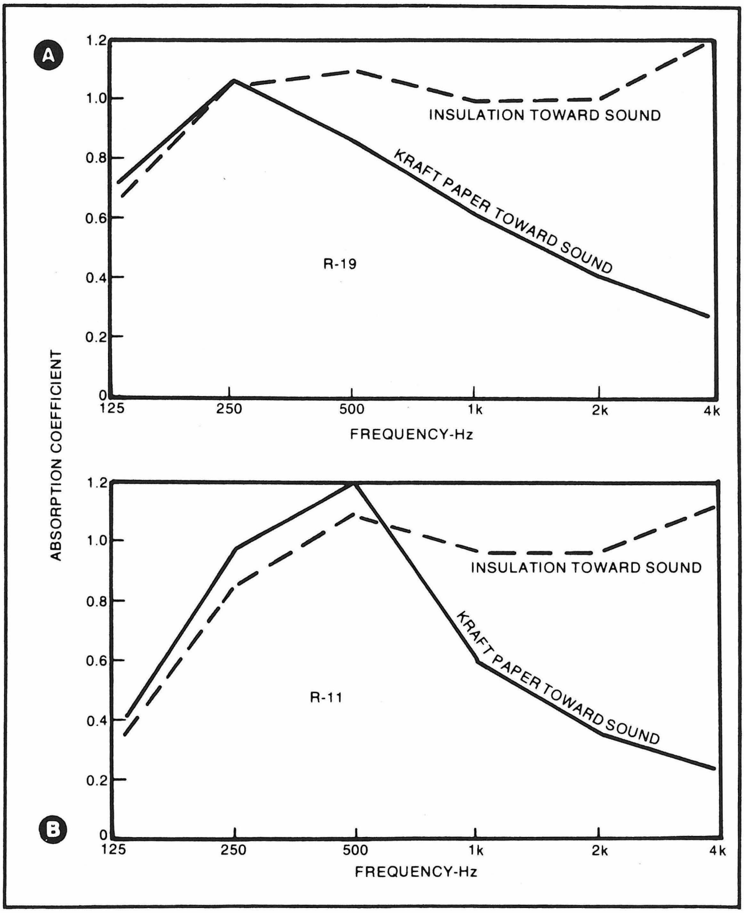


Fig. 9-9. When ordinary building insulation is used as a wall treatment (perhaps with fabric facing) the position of the kraft paper backing becomes important. (A) R-19 Fiberglass building insulation, (B) R-11 Fiberglass building insulation, Mounting 4.

Building insulation commonly comes with a kraft paper backing. Between walls this paper has no significant effect, but if building insulation is to be used as a sound absorber on walls, perhaps behind a fabric facing, the paper becomes significant. Figure 9-9 compares the sound absorption efficiency of R-19 (6 inch) and R-11 (3.5 inch) with the kraft paper backing exposed and with the glass fiber exposed to the incident sound. When the paper is exposed it shields the glass fiber from sound above 500 Hz but has little effect below 500 Hz. The net effect is an absorption peak at 250 Hz (R-19) and 500 Hz (R-11) which may be important in room treatment. With insulation exposed there is essentially perfect absorption above 250 Hz (R-19) or 500 Hz (R-11).

Building insulation has not caught on as an inexpensive absorbing material. One reason is that some sort of cosmetic and protective cover is required, but this is true of denser materials as well. Fabric, expanded metal, metal lath, hardware cloth, or even perforated vinyl wallcovering can be used as a cover. Do not be surprised by absorption coefficients greater than 1.0. This manufacturer elected to publish the coefficients actually obtained from laboratory measurements rather than arbitrarily reducing those over 1.0 to 0.99 or 1.0 as is the practice in the *Compendium* in the interests of uniformity of treatment of data. The greater absorption of the standard 8 × 9 ft sample results from edge diffraction and other effects which make the sample appear larger acoustically than it really is.

GLASS FIBER: BOARDS

This type of glass fiber usually used in the acoustical treatment of audio rooms is in the form of semi-rigid boards of greater density than building insulation. Typical of such materials are Owens-Corning Type 703 Fiberglas and Johns-Manville 1000 Series Spin-Glass, both of 3 lb/cu ft density. Other densities are available, for example Type 701 has a density of 1.58 lbs/cu ft and Type 705 a density of 6 lbs/cu ft. The Type 703 density, however, is widely applied in studios.

These semi-rigid boards of glass fiber do not excel cosmetically, hence they are usually covered with fabric. They do excel in sound absorption as shown in Fig. 9-10.

URETHANE FOAM

Although the 1980 edition of the *Compendium* lists 4½ pages of foams, there is much for vibration damping but little for Mounting 4.

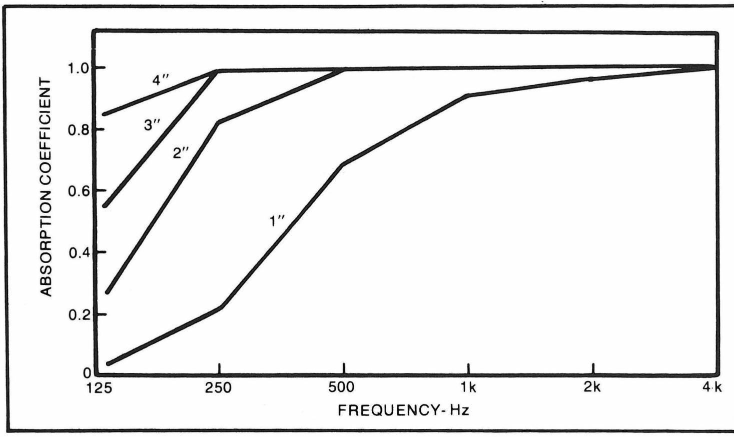


Fig. 9-10. Sound absorption characteristics of Owens-Corning Type 703 Fiberglas semirigid boards of 3 lbs/cu ft density and of various thicknesses. The Johns-Manville 1000 Series Spin-Glass boards have comparable absorption. (Data from the Compendium[1])

A foam product gaining in popularity in the treatment of audio rooms but not listed in the 1980 *Compendium* is Sonex, manufactured by Illbruck, USA, of Minneapolis, Minnesota, illustrated in Fig. 9-11. This material comes in 4 ft × 4 ft nested pairs, one sheet having wedges, the other matching indentations. While differing in form,

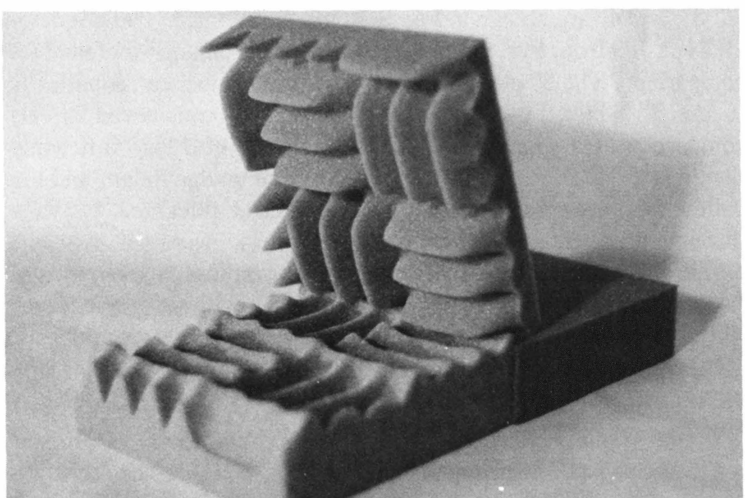


Fig. 9-11. Open cell urethane form contoured to simulate anechoic wedges. This material, called Sonex, is available in several thicknesses and colors. It is manufactured by Illbruck, USA, of Minneapolis, Minnesota.

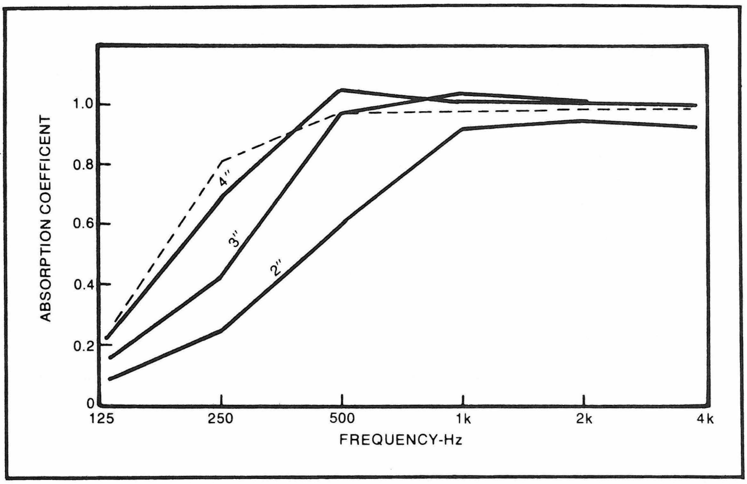


Fig. 9-12. Sound absorption characteristics of Sonex, of 2", 3", and 4" thickness, Mounting 4. The broken line is 2" Type 703 Fiberglas from Fig. 9-9 for comparison.

sheets of both types blend visually reasonably well and both have essentially the same absorption characteristics. In addition to the 4 ft × 4 ft sheets, Sonex is available as Audiotiles which are 15 inches square and 2 inches thick. Sonex has a density of 2 lbs/cu ft.

The sound absorption coefficients of Sonex for thicknesses of 2", 3", and 4" are shown in Fig. 9-12 for Mounting 4. The graph for 2" Type 703 from Fig. 9-10, shown as a broken line, is included for comparison. The 2" glass fiber is considerably superior acoustically to the 2" Sonex but a few things should be considered in this comparison. (1) The Type 703 has a density of 3 lbs/cu ft while Sonex is 2 lbs/cu ft. (2) The 2" Sonex is the wedge height and the valley thickness is far less, while the 703 thickness prevails throughout. (3) Comparing the two products is, in a sense, specious because the much higher cost of Sonex is justified in the minds of many by appearance and ease of mounting without fabric facing rather than straight acoustical considerations.

VICRACOUSTIC

Glass fiber is well established as an effective sound absorber at reasonable cost. The problem is that its appearance in the raw state leaves something to be desired and it requires protection from abrasion. Some of the most interesting and attractive panels of glass fiber are those manufactured by L.E. Carpenter and Company of Wharton, New Jersey. One typical panel is made up of a ⅜" compo-

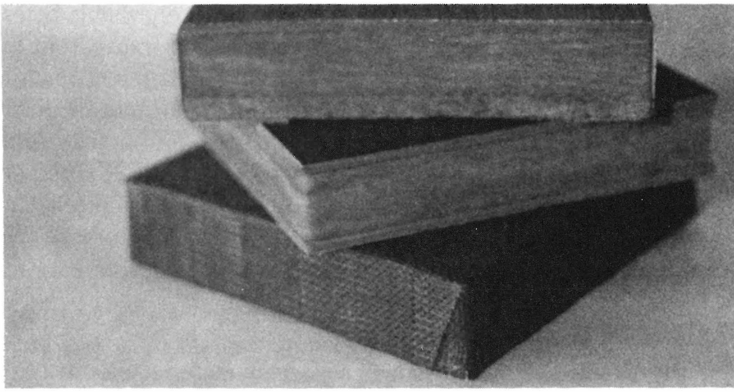


Fig. 9-13. The appearance of proprietary glass fiber panels offered by L.E. Carpenter and Company of Wharton, New Jersey. A molded ⅛″ panel of impact resistant glass fiber protects the 1″ or 2″ core of glass fiber absorbent. Attractive coverings of perforated vinyl or fabric dress up the panels.

sition backing board, a core of 1″ glass fiber, a protective facing of ⅛″ impact resistant molded glass fiber sheet, and an ornamental facing of perforated vinyl or fabric. Other types of panels eliminate the ⅜″ composition board backing or replace it with a second sheet of ⅛″ molded glass fiber. A 2″ glass fiber core is also available. These panels are made up at the factory to the customer's size and other specifications. Some 1700 colors and 100 patterns are available for the cosmetic facing. The construction of L.E. Carpenter panels is shown in Fig. 9-13. The sound absorption characteristics of panels with 1″ and 2″ glass fiber core thickness are shown in Fig. 9-14.

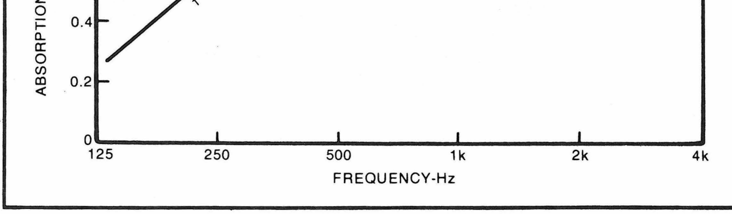


Fig. 9-14. Sound absorption characteristics of the L.E. Carpenter proprietary panels shown in Fig. 9-13. (Data from the Compendium[1])

The tough vinyl cover is perforated with scarcely visible holes to give about 18% perforation which is essentially transparent to sound. Panels may be hung by Z-clips, magnets, batten strips, or cement depending upon whether they are in the form of modular panels in wood or metal frames, flush-butted, or in unbroken monolithic form. The application of panels of the L.E. Carpenter type in studios and home listening rooms has been quite limited, but where good acoustical performance, durability, and appearance are given high priority, panels of this kind should be considered.

PARTICLE BOARD

Digressing briefly from sound absorbent materials, let us consider a high transmission loss material useful for doors. Mass is required of door material to reduce penetration of outside noise. One simple way of achieving mass is to use material such as that illustrated in Fig. 9-15. Solid core doors are available made of this material which is composed of a dense particle board core with hot-pressed board facings, total thickness 1¾″. The surface density of this material is about 5.3 lbs/sq ft. Making sound room doors of a favorable material such as this is an exercise in futility unless adequate weatherstripping is also installed and maintained properly.

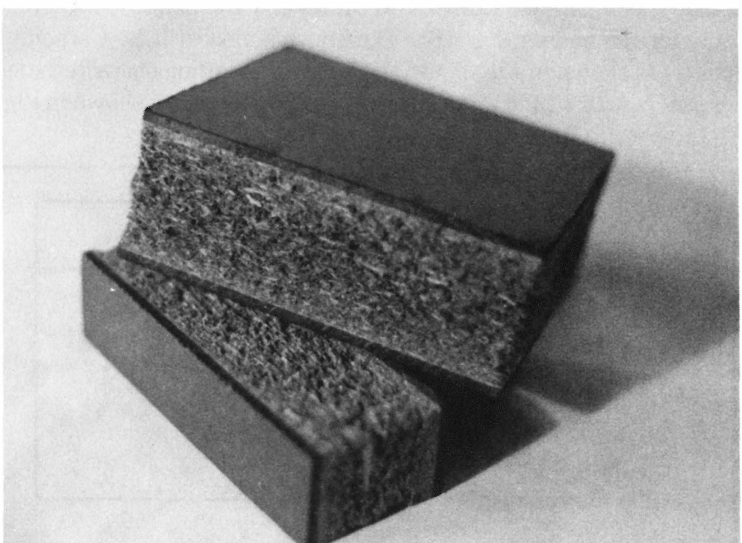


Fig. 9-15. Effective doors can be made of 1-¾″ solid core slabs of particle board faced with hot-pressed board.

PANEL ABSORBERS

In Chapter 1 we saw that a mass suspended from a spring vibrated at its natural frequency. Panels arranged with an air cavity behind act in a similar way. The mass of the panel and the springiness of the air in the cavity are resonant at some particular frequency. Sound is absorbed as the thin panel is flexed because of the friction of the fibers within the panel. The absorption of sound is maximum at the frequency at which the structure is resonant. This may be estimated from:

$$f_o = \frac{170}{\sqrt{(m)(d)}} \qquad \textbf{(9-1)}$$

in which

f_o = frequency, Hz
m = surface mass of the panel, lb per sq. ft. of panel surface
d = depth of air space, inches

To make it easy to use, equation 9-1 has been plotted in Fig. 9-16 for some commonly used materials. From the graph we see, for example, that ⅛″ plywood furred out from a solid wall by 2 × 4s on edge resonates at about 145 Hz. Masonite or other hardboard of ¼″ thickness and spaced out from the wall with 2 × 6s on edge resonates near 70 Hz.

Normally, flat panels used as acoustical elements would be spaced out from the wall with glass fiber in the cavity. This more than doubles the peak absorption coefficient. In Fig. 9-17 the absorption characteristics of three panel configurations are shown. In graph A is the simple case of 3/16″ plywood panels on 2″ battens.[8] If we refer to Fig. 9-16 and exercise a bit of judgment in reading midway between the ⅛″ and ¼″ plywood graphs, we estimate that this structure is resonant at about 175 Hz. The peak coefficient is about 0.3 which is about as high as one can expect for such structures. Graph B is for 1/16″ plywood with a 1″ glass fiber blanket and a ¼″ air space behind it.[9] Graph C is the same except for a ⅛″ panel. Note that the glass fiber filler has about doubled the peak absorption. The glass fiber has also shifted the peaks to a paint about 50 Hz lower than estimated from Fig. 9-16.

We have noted in Fig. 9-2 that the porous materials commonly show their greatest absorption in the high frequency region. The vibrating panel arrangements of Fig. 9-17 show their best absorption in the low frequencies. In treating small listening rooms and studios we shall find that structures giving good absorption in the lows are invaluable in controlling room modes.

Flat paneling can be highly decorative as well as effective in

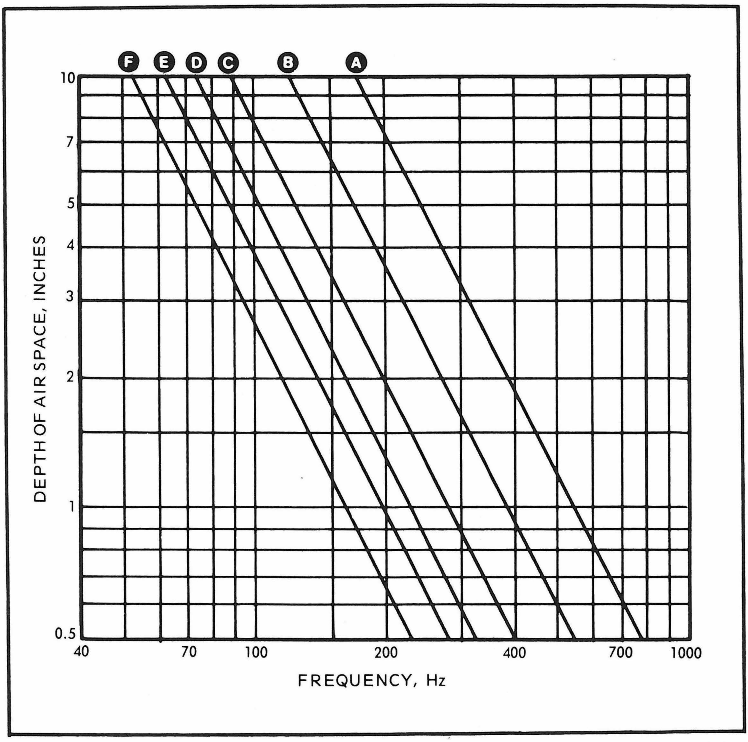


Fig. 9-16. A graphical presentation of Eq. 9-1 relating the surface mass of the panel, the depth of air space, and the frequency of resonance of the panel absorber.

(A) Surface mass = 0.1 lb/sq ft.
(B) Surface mass = 0.2 lb/sq ft.
(C) Surface mass = 0.37 lb/sq ft (1/8" plywood).
(D) Surface mass = 0.55 lb/sq/ft (1/8" hardboard).
(E) Surface mass = 0.74 lb/sq ft (1/4" plywood).
(F) Surface mass = 1.1 lb/sq ft (1/4" hardboard).

Example: 1/4" plywood with an air space of 3.75" resonates at about 100 Hz.

absorbing sound in the bass region. In fact, the excellent acoustics of some famous music rooms in Europe can be traced to the rich hardwood panels covering the walls.

POLYS: WRAPAROUND PANELS

Flat paneling in a room may brighten an interior decorator's eye and do some good acoustically, but wrapping a plywood or hardboard skin around some semicylindrical bulkheads can provide some very attractive features, as we have seen in Chapter 7. These polycylindrical elements (polys) are rather out of fashion now after a strong run of popularity in radio broadcasting, listening and record-

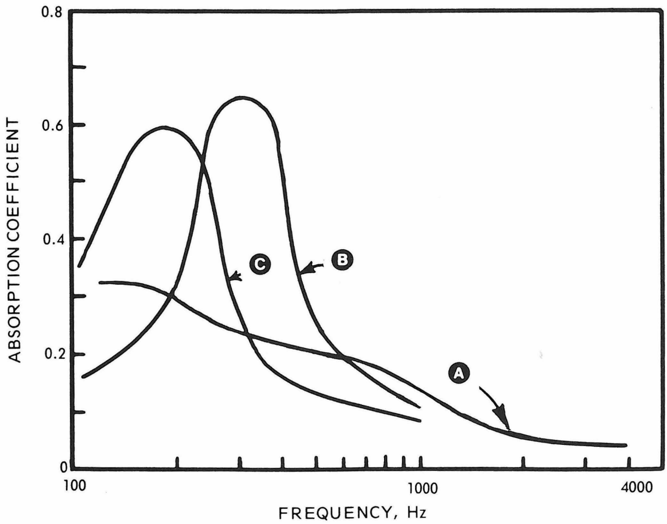


Fig. 9-17. Actual absorption measurements of three panel absorbers.
(A) 3/16" plywood with 2" air space[8].
(B) 1/16" plywood with 1" mineral wool and ¼" air space.[9]
(C) The same as (B) but for ⅛" panel[9].

ing studios, and motion picture scoring stages in the 1940s and 1950s. A few are used in top flight recording studios today. Visually, they are rather overpowering, which can be good or bad depending upon the effect one wants to achieve. With polys it is acoustically possible to achieve a good diffuse field along with liveness and brilliance, factors tending to oppose each other in rooms with flat surfaces.

One of the problems of using polys has been the scarcity of published absorption coefficients. The Russian acoustician, V. S. Mankovsky, has taken care of that in his book.[4] As expected, the larger the chord dimension, the better the bass absorption. Above 500 Hz there is little significant difference between the polys of different sizes.

The overall length of polys is rather immaterial, ranging in actual installations from the length of a sheet of plywood, to the entire length, width, or height of a studio. It is advisable, however, to break up the cavity behind the poly skin with randomly spaced bulkheads. The polys of Fig. 9-18 incorporate such bulkheads.

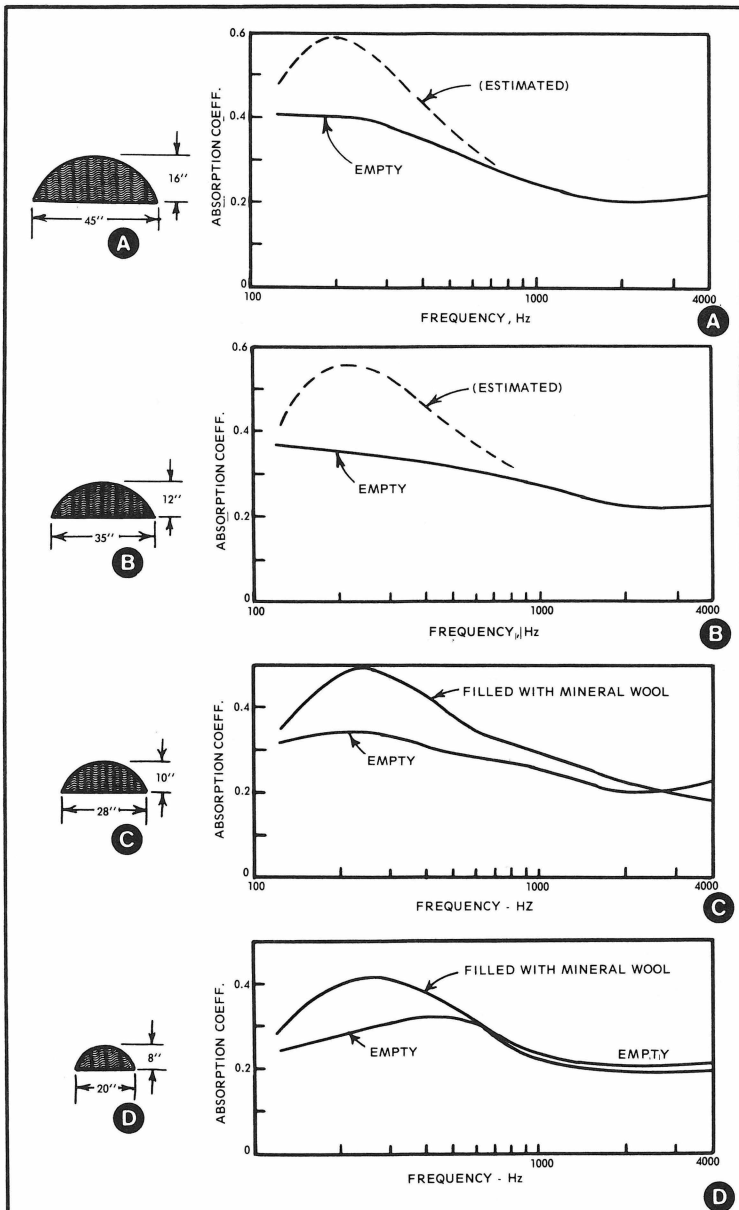


Fig. 9-18. Measured absorption of polycylindrical diffusers of various chord and height dimensions. In C and D, graphs are shown for both empty polys and for polys filled with mineral wool. In A and B only empty poly data are available; the broken lines are estimated absorption when filled with mineral wool. (After Mankovsky[4])

Should the polys be empty or filled with something? Mankovsky again comes to our rescue and shows us the effect of filling the cavities with absorbent material. Figures 9-18C and 9-18D show the increase in bass absorption resulting from filling the cavities with absorbent. If needed, this increased bass absorption may be easily achieved by simply filling the polys with glass fiber. If the bass absorption is not needed the polys may be used empty. The great value of this adjustable feature will become more apparent as we get into the actual acoustical design of listening rooms and studios in coming chapters.

POLY CONSTRUCTION

The construction of polycylindrical diffusers is reasonably simple.[10] A framework for vertical polys is shown in Fig. 9-19 mounted above a structure intended for a low frequency slat absorber. In this photograph the variable chord dimensions are appar-

Fig. 9-19. The construction of polys in a motion picture sound mixing studio. Note the foam rubber anti-rattle strip on the edge of each bulkhead. Also note the random spacing of the bulkheads. (Moody Institute of Science photo, reprinted with permission of the Journal of the Audio Engineering Society[10].)

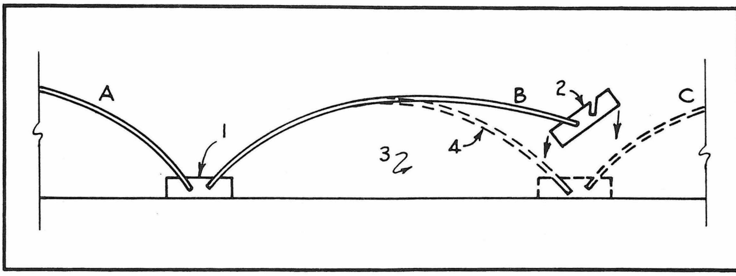


Fig. 9-20. The method of stretching the plywood or hardboard skin over the poly bulkheads shown in Fig. 9-18.

ent and also the random placement of bulkheads so that cavities will be of various volumes resulting in various natural cavity frequencies. It is desirable that each cavity be essentially air tight, isolated from adjoining cavities by well-fitted bulkheads and framework. Irregularities in the wall can be sealed with a nonhardening acoustical sealant. The bulkheads of each poly are carefully cut to the same radius on a bandsaw. Sponge rubber weatherstripping with an adhesive on one side is struck to the edge of each bulkhead to insure a tight seal against the plywood or hardboard

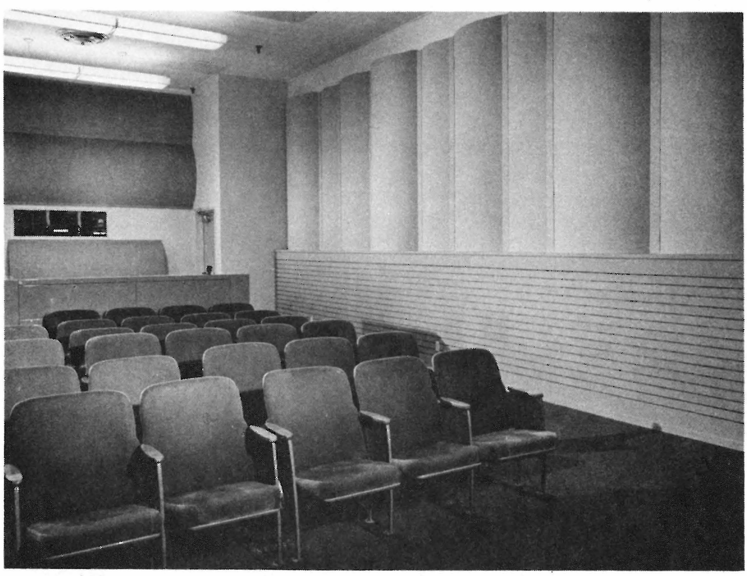


Fig. 9-21. Finished poly array of Fig. 9-19 mounted on the wall above a low frequency absorber structure. Note that the axes of the polys on the rear wall are perpendicular to the axes of the polys on the other wall. (Moody Institute of Science photo, reprinted with permission of the Journal of the Audio Engineering Society[10].)

cover. If such precautions are not taken, annoying rattles and coupling between cavities can result.

The polys of Fig. 9-19 at Moody Institute of Science utilize ⅛″ tempered Masonite as the poly skin. A few hints can simplify the job of stretching this skin. In Fig. 9-20 slots of a width to fit the Masonite snugly are carefully cut along the entire length of strips 1 and 2 with a radial saw. Let us assume that poly A is already mounted and held in place by strip 1 which is nailed or screwed to the wall. Working from left to right, the next job is to mount poly B. First the left edge of Masonite sheet B is inserted in the remaining slot of strip 1. The right edge of Masonite sheet B is then inserted in the left slot of strip 2. If all measurements and cuts have been accurately made, swinging strip 2 against the wall should make a tight seal over the bulkheads 3 and weatherstripping 4. Securing strip 2 to the wall completes poly B. Poly C is mounted in a similar fashion and so on to the end of the series of polys. The end result is shown in Fig. 9-21. Notice that the axes of symmetry of the polys on the side wall are perpendicular to those on the rear wall. If polys were used on the ceiling, their axes should be perpendicular to both the others.

It is quite practical and acceptable to construct each poly as an entirely independent structure rather than building them on the wall. Such independent polys can be spaced at will.

PERFORATED PANEL ABSORBERS

Perforated hardboard or plywood panels spaced from the wall constitute a resonant type of sound absorber.[11,12,13,14] Each hole acts as the neck of a Helmholtz resonator and the share of the cavity behind "belonging" to that hole is comparable to the cavity of the Helmholtz resonator. In fact, we can view this structure as a host of coupled resonators. If sound arrives perpendicular to the face of the perforated panel, all the tiny resonators are in phase. For sound waves striking the perforated board at an angle, the absorption efficiency is somewhat decreased. This loss can be minimized by sectionalizing the cavity behind the perforated face with an egg crate type of divider of wood or corrugated paper.

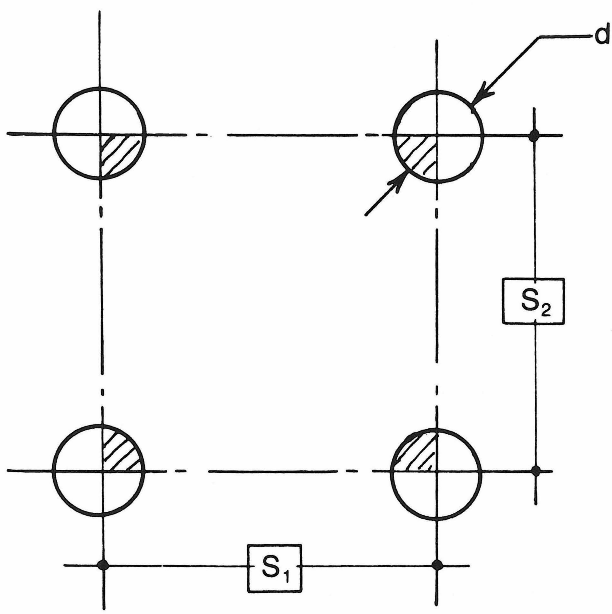
The frequency of resonance of perforated panel absorbers backed by a subdivided air space is given approximately by:

$$f_o = 200 \sqrt{\frac{p}{(d)\,(t)}} \qquad \textbf{(9-2)}$$

in which

f_o = frequency, Hz
p = perforation percentage,
= hole area divided by panel area × 100
t = effective hole length, inches, with correction factor applied,
= (panel thickness) + (0.8) (hole diameter)
d = depth of air space, inches

There is a certain amount of confusion in the literature concerning p, the perforation percentage. Some writers use the ratio of hole area to panel area, rather than the percentage of hole area to panel area, introducing an uncertainty factor of 100. This perforation percentage is easily calculated by reference to Fig. 9-22.



$$\text{Perforation } \% = \frac{\pi \left(\frac{d}{2}\right)^2}{(S_1)\,(S_2)} \times 100$$

Fig. 9-22. Method of calculating the percent perforation, p, of Eq. 9-2.

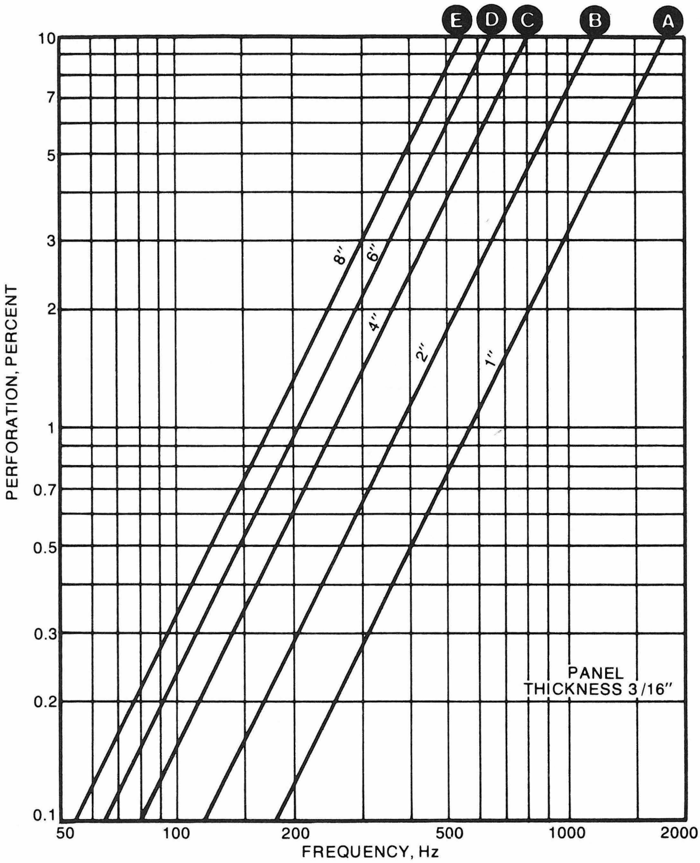


Fig. 9-23. A graphic presentation of Equation 9-2 relating percent perforation of perforated panels, the depth of air space, and the frequency of resonance. These graphs are for a panel of 3/16″ thickness.

(A) For 1″ furring lumber. The lines are drawn to correspond to furring lumber which is finished, e.g., the line for 8″ is actually 7-¾″ air space.
(B) For 2″ furring lumber.
(C) For 4″ furring lumber.
(D) For 6″ furring lumber.
(E) For 8″ furring lumber.

Equation 9-2 is true only for circular holes. This information is presented in graphical form in Fig. 9-23 for a panel thickness of 3/16″. Common pegboard with holes 3/16″ in diameter spaced 1″ on centers with the configuration of Fig. 9-22 has 2.75% of the area in holes. If this pegboard is spaced out from the wall by 2 × 4s on

edge, the system resonates at about 420 Hz and the peak absorption appears near this frequency.

In commonly available perforated materials, such as pegboard, the holes are so numerous that resonances at only the higher frequencies can be obtained with practical air spaces. To obtain much needed low frequency absorption you may have to drill your own. Drilling 7/32″ holes 6″ on centers gives a perforation percentage of about 1.0%. With zero perforation percentage you are right back to the solid panels of Eq. 9-1.

Figure 9-24 shows the effect of varying hole area from 0.18% to 8.7% in a structure of otherwise fixed dimensions. The plywood is 5/32″ thick perforated with 3/16″ holes except for the 8.7% case in which the hole diameter is about ¾″. The perforated plywood sheet

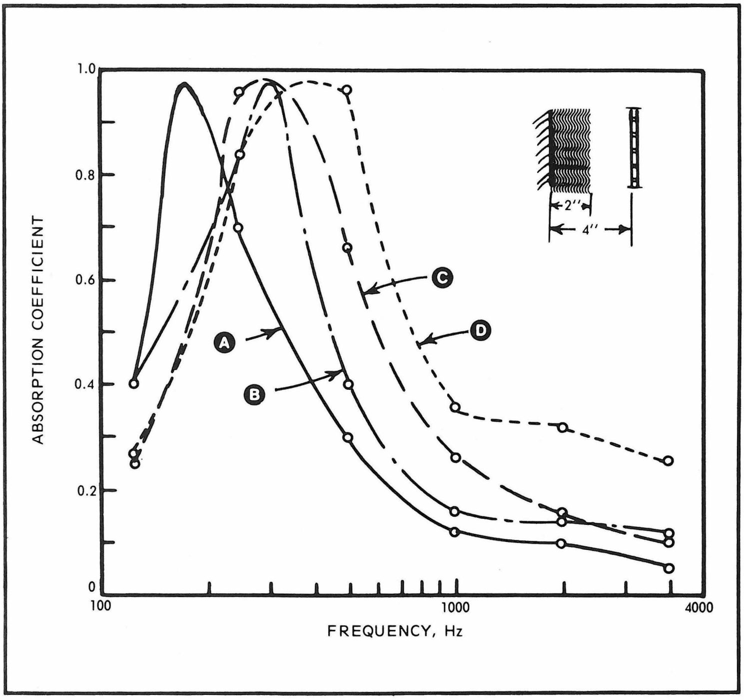


Fig. 9-24. Actual absorption measurements of perforated panel absorbers of 4″ air space, half filled with mineral wool and for panel thickness of 5/32″.

(A) Perforation 0.18%.
(B) Perforation 0.79%.
(C) Perforation 1.4%.
(D) Perforation 8.7%.

Note that the presence of the mineral wool shifts the frequency of resonance considerably from the theoretical values of Equation 9-2 and Fig. 9-23. (Data from Mankovsky[4].)

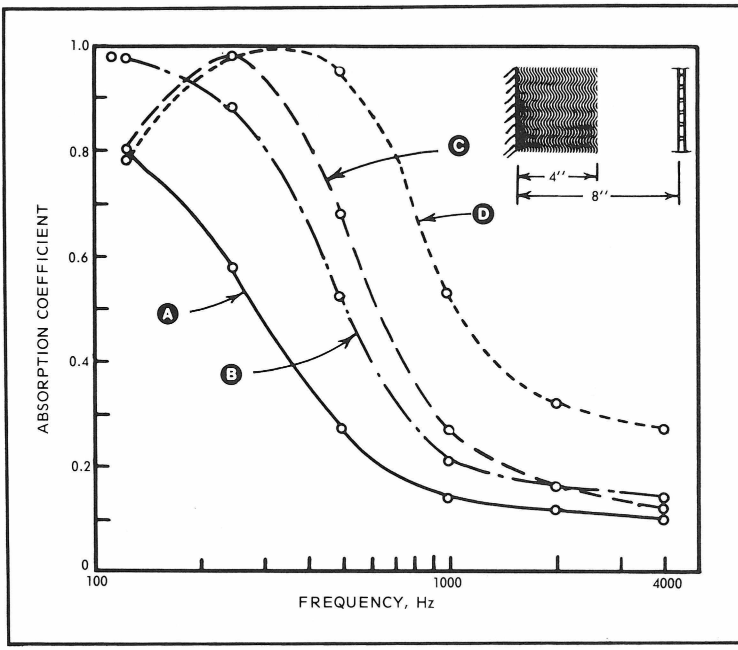


Fig. 9-25. Actual absorption measurements on the same perforated panel absorbers of Fig. 9-24 except that the air space is increased to 8", half of which is taken up with mineral fiber. Panel thickness is 5/32".

(A) Perforation 0.18%.
(B) Perforation 0.79%.
(C) Perforation 1.4%.
(D) Perforation 8.7%.

(Data from Mankovsky[4]).

is spaced 4" from the wall and the cavity is half filled with glass fiber and half is air space.[4]

Figure 9-25 is identical to Fig. 9-24 except that the perforated plywood is spaced 8" and glass fiber of 4" thickness is mounted in the cavity. The general effect of these changes is a substantial broadening of the absorption curve.

It would be unusual to employ such perforated panel absorbers without acoustic resistance in the cavity in the form of glass fiber batts or boards, Without such resistance the graph is very sharp. One possible use of such sharply tuned absorbers would be to control troublesome room modes or isolated groups of modes with otherwise minimum effect on the signal and overall room acoustics.

SLAT ABSORBERS

Another type of resonant absorber is that utilizing closely

spaced slats over a cavity.[15] The mass of the air in the slots between the slats reacts with the springiness of air in the cavity to form a resonant system, again comparable to the Helmholtz resonator. The glass fiber board usually introduced behind the slots acts as a resistance, broadening the peak of absorption. The narrower the slots and the deeper the cavity, the lower the frequency of maximum absorption.

The resonance frequency of the slat absorber can be estimated from the statement:[15]

$$f_o = 2160 \sqrt{\frac{2r}{(d)\,(D)\,(w+2r)}} \qquad \textbf{(9-3)}$$

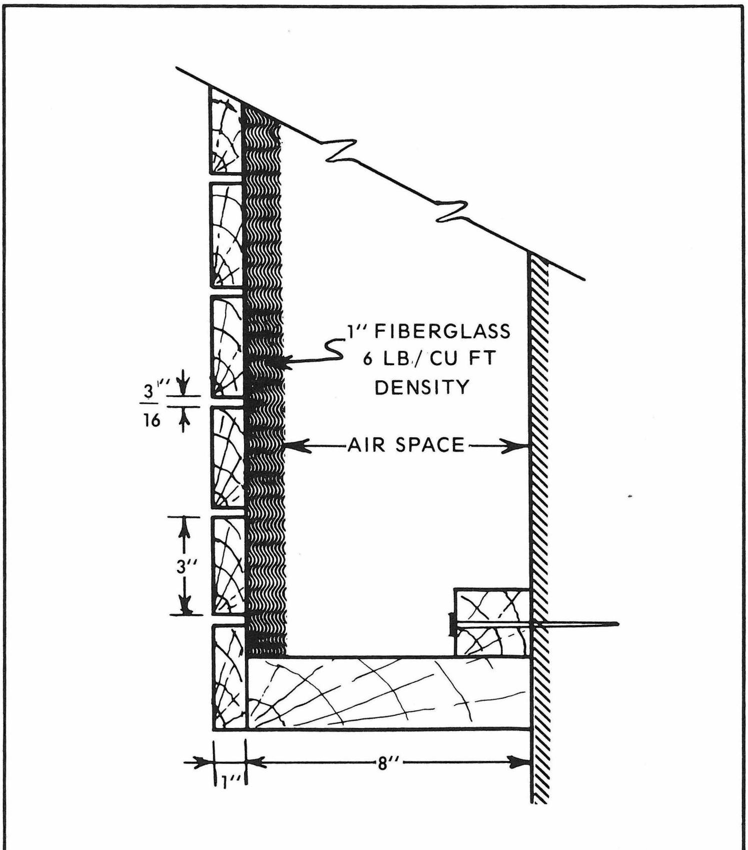


Fig. 9-26. A typical slat absorber giving peak absorption at about 150 Hz.

in which

$$D = \text{depth of air space, inches}$$
$$d = \text{thickness of slat, inches}$$
$$w = \text{width of slat, inches}$$
$$2r = \text{width of slot, inches}$$

For example, let us take the typical slat absorber pictured in Fig. 9-26 in which $D = 8''$, $d = 1''$, $w = 3''$, and $2r = \frac{3''}{16} = 0.1875''$. Substituting these values in Eq. 9-3 gives:

$$f_o = 2160 \sqrt{\frac{0.1875}{(1)\,(8)\,(3 + 0.1875)}}$$
$$= 185 \text{ Hz.}$$

Suggestion: if slats are mounted vertically, it is recommended that they be finished in a dark color conforming to the shadows of the slots to avoid some very disturbing "picket fence" optical effects!

MODULES

The British Broadcasting Corporation has pioneered a modular approach to the acoustical treatment of their numerous small voice studios which is very interesting.[16] Because they have applied it in several hundred such studios economically and with very satisfactory acoustical results, it deserves our critical attention. Basically, the idea is to cover the walls with standard-sized modules, say 2 × 3 ft, having a maximum depth of, perhaps, 8″. These can be framed on the walls to give a flush surface appearing very much like an ordinary room or they may be made into attractive boxes with grill cloth covers mounted on the walls in regular patterns. All modules may be made to appear identical, but the similarity is only skin deep.

There are commonly three, or perhaps four, different types of modules, each having its own distinctive contribution to make acoustically. Figure 9-27 shows the radically different absorption characteristics obtained by merely changing the covers of the standard module. This is for a 2 × 3 ft module having a 7″ air space and a 1″ semirigid glass fiber board of 3 lb/cu ft density inside. The wideband absorber has a highly perforated cover (25% or more perforation percentage) or no cover at all, yielding essentially complete absorption down to about 200 Hz. Even better low-frequency absorption is possible by breaking up the air space with egg crate type dividers of corrugated paper to discourage unwanted

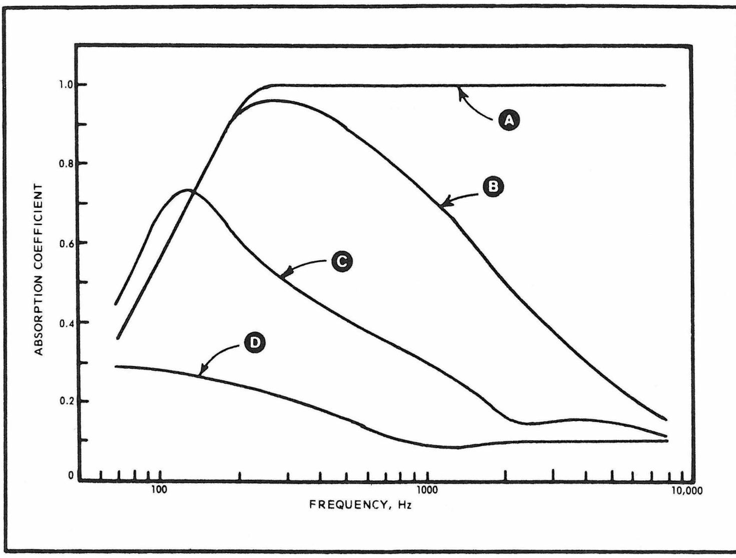


Fig. 9-27. Modular absorber having a 7" air space and 1" semiridged glass fiber board of 9 to 10 lb/cu ft density behind the perforated cover.
(A) No perforated cover at all, or at least more than 25% perforation.
(B) 5% perforated cover.
(C) 0.5% perforated cover.
(D) ⅜" plywood cover, essentially to neutralize the module.
(Data from Brown[16].)

resonance modes. A cover ¼" thick with a 5% perforation percentage peaks in the 300-400 Hz range. A true bass absorber is obtained with a low perforation cover (0.5% perforation). If essentially neutral modules are desired, they can be covered with ⅜" and ¼" plywood which would give relatively low absorption with a peak around 70 Hz. Using these three or four modules as acoustical building blocks, the desired effect can be designed into a studio by specifying the number of distribution of each of the basic types.

Figure 9-28 shows an adaptation from BBC practice where the wall is used as the "bottom" of the module box. In this case the module size is 2 × 4 ft. The modules are fastened to the 2 × 2" mounting strips which, in turn, are fastened to the wall. A studio wall 10 ft high and 23 or 24 ft long might use 20 modules of distributed types, four modules high and five long. It is good practice to have acoustically dissimilar modules opposing each other on opposite walls.

The question that comes to mind is, "How about diffusion of sound with such modular treatment?" BBC experience has shown

that careful distribution of the different types of modules results in adequate diffusion.

PLACEMENT OF MATERIALS

The application of sound absorbing materials in random patches has already been mentioned as an important means of achieving diffusion. Other factors than diffusion might influence placement. If several types of absorbers are used, it is desirable to place some of each type on ends, sides, and ceiling so that all three axial modes (longitudinal, transverse, and vertical) will come under their influence. In rectangular rooms it has been demonstrated that absorbing material placed near corners and along edges of room surfaces is most effective. In speech studios, some absorbent effective at the higher audio frequencies should be applied at head height on the walls. In fact, material applied to the lower portions of high walls may be as much as twice as effective as the same material placed elsewhere. It is rather an obvious conclusion that untreated surfaces should never face each other.

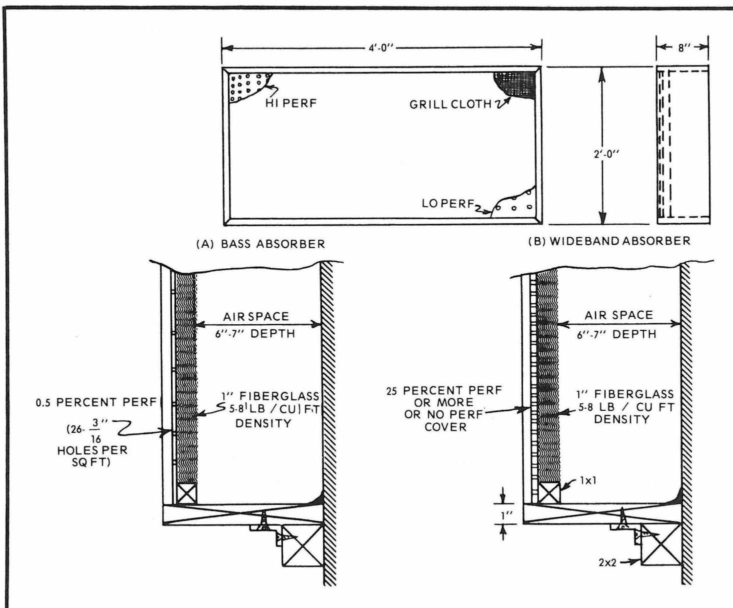


Fig. 9-28. Plan for a practical module absorber utilizing the wall as the bottom of the module.
(Left) Bass absorber.
(Right) Wideband absorber.

Winston Churchill once remarked that as long as he had to wear spectacles he intended to get maximum cosmetic benefit from them. So it is with placement of acoustical materials. After the demands of acoustical function have been met, every effort should be made to arrange the resulting patterns, textures, and protuberances into esthetically pleasing arrangements, but do not let your priorities be reversed!

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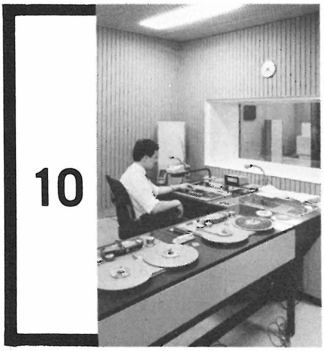

The Decay of Sound in Small Rooms

If you push the accelerator pedal of your automobile down to a certain point and hold it there, the car accelerates to a certain speed, and if the road is smooth and level this speed will remain constant. With this accelerator setting the engine produces just enough torque to overcome all the frictional losses and a balanced (steady-state) condition results. So it is with sound in a room. A loudspeaker is arranged to emit white noise into a room. As the switch is closed the sound quickly builds up to a certain level as shown by Fig. 10-1. This is the steady-state or equilibrium point at which the sound energy radiated from the loudspeaker is just enough to supply all the losses at the boundaries of the room. If the switch is opened, it takes a finite length of time for the sound level to decay to inaudibility. This "hanging-on" of the sound in a room after the exciting signal has been removed is called reverberation and it has a very important bearing on the acoustical quality of the room.

Recordings made outdoors or in an anechoic (echo-free) room have a dead quality. Sound may spread in all directions from the source but only that traveling toward the microphone will be recorded under these circumstances. If the same source, microphone, and recording system are taken indoors, reflections from the walls, ceiling, and floor give the recorded sound quite a different quality. Controlled reverberation, whether natural or artificial, can enhance the quality of voice and music being recorded or reproduced but too

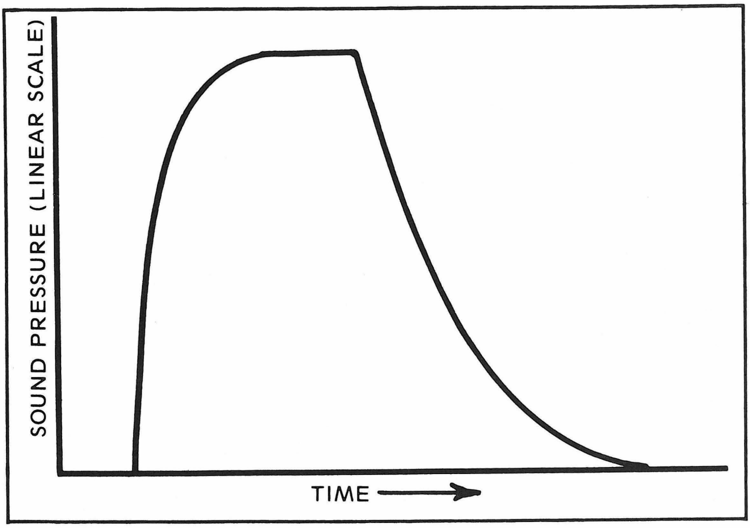

Fig. 10-1. Rise and decay of sound in a room. The linear sound pressure scale yields the familiar exponential decay shape.

much or too little reverberation can create problems. In the design of small listening rooms or studios, reverberation must be considered along with diffusion of sound and the intrusion of unwanted sounds from the outside.

REVERBERATION TIME

Reverberation time is arbitrarily defined as that time required for a sound to die away to one-thousandth of its initial sound pressure. As this corresponds to a drop in sound pressure level of 60 dB, it is convenient to abbreviate reverberation time to T_{60}. This definition is related to the characteristics of the human ear in that it represents very approximately a decay from a fairly loud listening level to the threshold of audibility in the presence of normal environmental noise, say from 85 phons to 25 phons in Fig. 4-4. In fact, a stopwatch, a sound source (organ pipes were used historically), and a keen ear can be used to estimate reverberation times of 1 second or longer, found in the larger enclosures.

As sound decays in a room it follows what the mathematician calls an exponential curve. Plotted on a linear scale the exponential decays as shown in Fig. 10-1. The same thing plotted as sound pressure level in dB, which is a logarithmic scale, is shown in Fig. 10-2. On a logarithmic pressure scale the exponential decay becomes a straight line which is more convenient to handle. On such a

straight-line decay the meaning of reverberation time is as indicated in Fig. 10-2.

REVERBERATION AND NORMAL MODES

Our introduction to the measurement of reverberation will be through a method wholly unsuited to practical use, but which, as an investigative tool, reveals some very important factors which focus attention on the normal modes of rooms. Historically, in Broadcasting House in New Delhi, India, is (or was) a Studio 10 used for news broadcasts. Measurements of reverberation time in this studio were reported by Beranek[1] and later analyzed by Schultz. The first set of measurements were made in the completely bare, untreated room. Knowing something of construction in India, it can be safely assumed that concrete and ceramic tile dominated the room surfaces. The measurements were made with sine wave signals and great patience and care were exercised to obtain the detailed results.

The reverberation time graph of Fig. 10-3 is not that of the New Delhi studio, but similar to it. Changes have been made below 100 Hz to simulate the measured peaks from New Delhi, but they have been redrawn to fit our studio example (23.3 × 16 × 10 ft) studied in detail in Chapter 6 and illustrated in Table 6-3 and Fig.

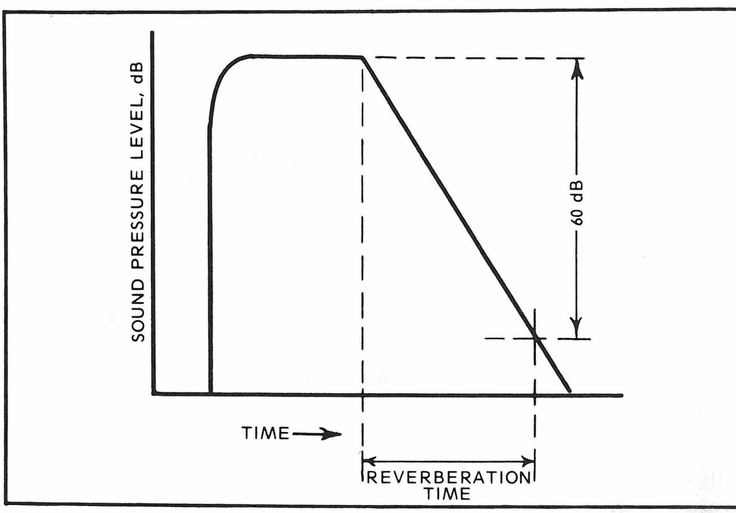

Fig. 10-2. Rise and decay of sound in a room with a logarithmic vertical scale, sound pressure level in dB. The exponential decay is now a straight line, convenient for measuring reverberation time which is defined as the time required for sound to decay 60 dB.

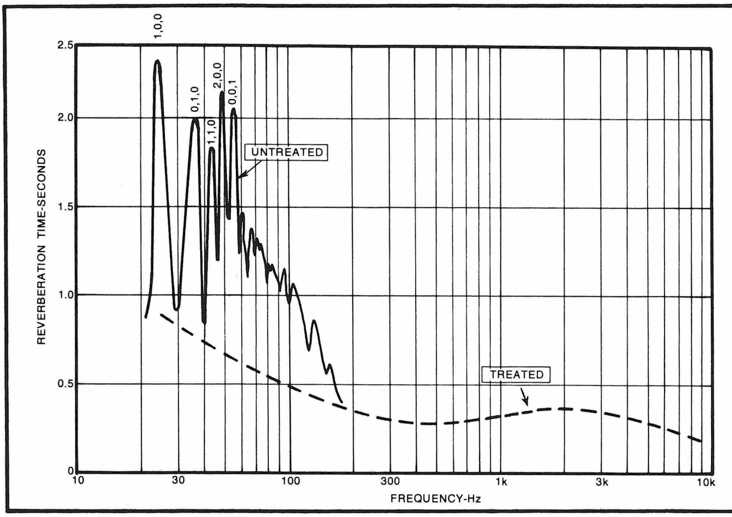

Fig. 10-3. Reverberation time measured with pure sine signals at low frequencies reveals slow sound decay (long reverberation time) at the modal frequencies. These peaks apply only to specific modes and are not representative of the room as a whole. High modal density, resulting in uniformity of distribution of sound energy and randomizing of directions of propagation, is necessary for reverberation equations to apply. (Beranek[1] and Schultz[2].)

6-4. We shall pay particular attention to the lowest five modal frequencies.

Starting with the oscillator set to about 20 Hz, below the first axial mode, the acoustics of the room do not load the loudspeaker

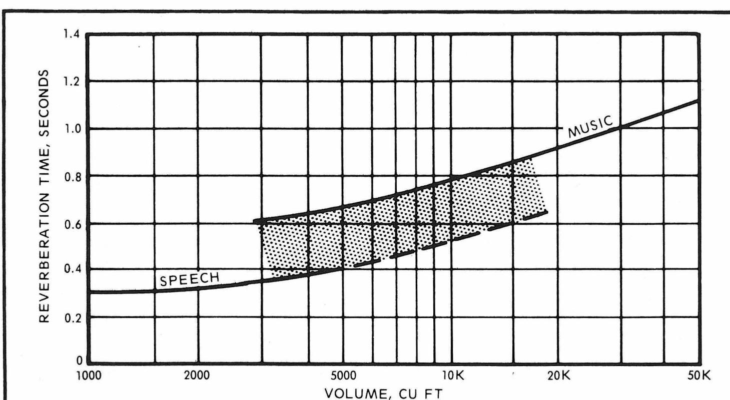

Fig. 10-4. Optimum reverberation time for small listening rooms and studios as a function of room volume. The shaded area between the speech and music curves may be considered a compromise region for studios to be used for both speech and music.

and a relatively weak sound is produced with the amplifier gain turned up full (even assuming the use of a good sub-woofer). As the oscillator frequency is adjusted upward, however, the sound becomes very loud as the 1,0,0 mode (24.18 Hz) is energized. Slowly adjusting the oscillator upward we go through a weak valley but at the frequency of the 0,1,0 mode (35.27 Hz) there is high level sound once more. Similar peaks are found at the 1,1,0 tangential mode (42.76 Hz), the 2,0,0 axial mode (48.37 Hz), and the 0,0,1 axial mode (56.43 Hz).

Now that the loudness of peaks and valleys have been explored, let's examine the decay of sound. Exciting the 1,0,0 mode at 24.18 Hz, the decay is measured as the source is interrupted and we get a long reverberation time of 2.3 seconds. We observe similar slow decays at 35.27 Hz, 42.76, 48.37, and 56.43 Hz with faster decays (shorter reverberation times) in between. The decays at the modal frequencies are decay rates characteristic of individual modess, not of the room as a whole.

What is going on here? Long reverberation time implies low absorbence and short reverberation time implies high absorbence. It is difficult to believe that the sound absorbing qualities of the walls, floor, and ceiling vary this much within a frequency range of a few hertz. For the 1,0,0 mode, only the absorbence of the two ends of the room come into play, the four other surfaces are not involved at all. For the 0,0,1 mode, only the floor and ceiling are involved. All we have done in this low frequency range is to measure the decay rate of individual modes, definitely not the average condition of the room.

We see now why there is that big question mark over applying the concept of reverberation time to small rooms having dimensions comparable to the wavelength of sound. Schultz states[2] that reverberation time is a statistical concept "in which much of the mathematically awkward details are averaged out". In small rooms these details are not averaged out.

The reverberation time formulas of Sabine, Eyring and others are based on the assumption of an enclosed space in which there is highly uniform distribution of sound energy and random direction of propagation of the sound. At the low frequency points of Fig. 10-3 energy is distributed very unevenly and direction of propagation is far from random. After the room was treated, reverberation time measurements followed the broken line, but statistical randomness still does not prevail below 200 Hz even though modal frequencies are brought under some measure of control.

DECAY RATE

The definition of reverberation time is based on uniform distribution of energy and random directions of propagation. Because these conditions do not exist in small rooms, there is some questions as to whether what we measure should be called reverberation time. It is more properly termed "decay rate". A reverberation time of 0.3 second is equivalent to a decay rate of 60 dB/0.3 sec. = 200 dB per second. The use of decay rate instead of reverberation time would tell the experts that we are aware of the basic problems, but little else. Speech and music sounds in small rooms do decay even though the modal density is too low to hang the official "reverberation time" tag on the process.

THE SABINE COMPROMISE—HOW BAD IS IT?

Measuring decay rate of single modes can give us spurious and highly variable results as shown in Fig. 10-3. The reverberation time results improve as modal density increases. Let us abandon the sine wave signal method which was instructive in Fig. 10-3 and consider the use of bands of random noise. Table 10-1 lists the octave bands commonly used and a calculated approximation of the number of modes in each band for our 23.3 × 16 × 10 ft room. Table 6-3 lists the actual modes spanning the first two octaves and by actual count we get 13 modes for the 63 Hz octave and 56 for the 125 Hz octave which compares roughly with the 13 and 77 of Table 10-1. The 63 Hz octave has the fewest modes with 4 axial, 7 tangential, and 2 oblique which gives a rather lean sampling. Each of the 13 modes has a bandwidth of about 6 or 7 Hz which result in fair coupling of the 13 modes. So, from the standpoint of modal density,

Table 10-1. Number of Modes in Octave Bands.

Room: 23.3 × 16 × 10 ft

Octave Band Center Freq. Hz (f)	Octave Band Edges Lower $f/\sqrt{2}$	Octave Band Edges Upper $(f)(\sqrt{2})$	Approximate Number of Modes/Octave (calc)	Average Spacing Modes/Hz
62.5	44.2	88.4	13.	0.3
125	88.4	176.8	77.	0.9
250	176.8	353.6	513.	2.9
500	353.6	707.1	3,700.	10.5
1k	707.1	1,414.	28,300.	40.
2k	1,414.	2,828.	221,000.	156.
4k	2,828.	5,657	1,750,000.	617
8k	5,657	11,314	13,900,000.	2,450.

the lowest (worst) octave measurement band is marginal, but really not too bad. Further, reverberation measurements are often omitted for the 63 Hz octave because of great fluctuations during the decay period which contributes to inaccuracy in determining the decay slope.

Octave bands at successively higher frequencies have a dramatic increase in the number of modes. In fact, the 4 kHz octave band has almost 2 million modes in it or about 600 modes per Hz of bandwidth! This should provide considerable statistical distribution of angles of incidence and averaging out of other variables characterizing normal modes.

REVERBERANT FIELD

In our 23.3 × 16 × 10 ft room the volume is 3,728 cu ft. The inner surface area is 1,533 sq ft. By statistical theory (geometrical ray acoustics on which the Sabine equation is based) the mean free path (the average distance sound travels between reflections) is 4V/S or (4) (3728)/1533 = 9.7 ft. If the reverberation time is 0.3 second, there would be at least 35 reflections during the 60 dB decay. This would appear to be a fair involvement of all room surfaces.

In a small, relatively dead room such as the average studio, control room, and listening room, one never gets very far away from the direct influence of the sound source. A true reverberant field is often below the ambient noise level. The reverberation time equations, soon to be considered, have been derived for conditions that exist only in the reverberant field. In this sense, then, the concept of reverberation time is inapplicable to small, relatively dead rooms. And yet we measure something that looks very much like what is measured in large, more live spaces. What is it? What we measure is the decay rate of the normal modes of the room.

Each axial mode decays at its own rate determined by the absorbence of a pair of walls and their spacing. Each tangential and oblique mode has its own decay rate determined by distance traveled, the number of surfaces involved, the variation of the absorption coefficient of the surfaces with angle of incidence, etc. Measuring the decay rate of the 125 Hz octave in the room of Table 10-1 yields the average rate of decay of 5 axial, 26 tangential, and 25 oblique modes. Whatever average decay rate is measured for this octave for random noise will surely be representative of the average decay rate at which that octave of speech or music signals would die away. Although the applicability of computing reverberation time

from the equations based on reverberant field conditions might be questioned because of the lack of reverberant field, the measured decay rates (by whatever name you call them) most certainly apply to this space and to these signals.

ACOUSTICAL DESIGN PRINCIPLES FOR SMALL ROOMS

The acoustical design of rooms still must follow rather traditional patterns, even if the rooms are small. For example, in designing a small voice studio we would give attention first to the size of the room and then the proportions of the room to give the best distribution of modal frequencies at low frequencies. This is insurance against voice colorations due to unequal spacing and piling up of modal frequencies. The space must also be protected from interfering noise. The reverberatory characteristics must then be brought under control as absorbents are applied to the surfaces of the room. Even if Sabine's classical reverberation equation is based on thoroughly mixed sound fields, unattainable, in any complete sense, in a small room, we use it tentatively in this initial design stage because it has proved to be a helpful tool in reaching an acceptable balance between the wholly unacceptable extremes of totally dead and totally live conditions. We can do this with full awareness of the compromises involved and end up asking only one question, "Does the room sound right?".

OPTIMUM REVERBERATION TIME

Too long a reverberation time leads to confusion of consecutive syllables in speech with a resulting loss in intelligibility and slurring of phrases in music. Too short a reverberation time results in a "dead" effect and loss of brilliance. This would lead one to suspect that there is some optimum reverberation time in between. Figure 10-4 represents an attempt to summarize the acoustical "wisdom of the ages" in regard to optimum reverberation time as a function of room volume. As expected, the optimum reverberation time increases with the size of the listening room or studio. A small studio to be used for both speech and music can be designed around the shaded area of Fig. 10-4 which represents a modest compromise.

Figure 10-4 appears to be clear-cut and definite. It is anything but that! It represents the author's subjective judgment of the subjective judgments of scores of authorities who do not agree among themselves on the optimum reverberation time. Because of this highly uncertain factor there is little sense in straining to achieve a reverberation time of 0.367 seconds when we are not even

sure whether it should be 0.3 or 0.4. However, we can be assured that as far as reverberation times are concerned, following Fig. 10-4 will result in reasonably optimized and very usable conditions.

Optimum reverberation for speech lies close to the single curve in Fig. 10-4. For music the optimum curve should be a fuzzy line instead of a sharp line because the optimum reverberation time for music depends upon the kind of music. Fast, light, intricate music generally requires a shorter reverberation time than the broader, more ponderous music. The shaded area of Fig. 10-4 represents a compromise area for combination listening rooms or studios.

Many words have been written on how reverberation time should vary with frequency. Fortunately for us, there is now much better agreement on this subject. Uniform reverberation time as frequency is varied means that all components of the signal die away at the same rate. For the smaller listening rooms and studios at least, a safe first approach would be to make the reverberation time essentially constant throughout the audible spectrum.

BASS RISE OF REVERBERATION TIME

The goal in voice studios is to achieve a reverberation time that is the same throughout the audible spectrum. This may be difficult to realize, especially at low frequencies. Adjustment of reverberation time at high frequencies is easily accomplished by adding or removing relatively inexpensive absorbers. At low frequencies the

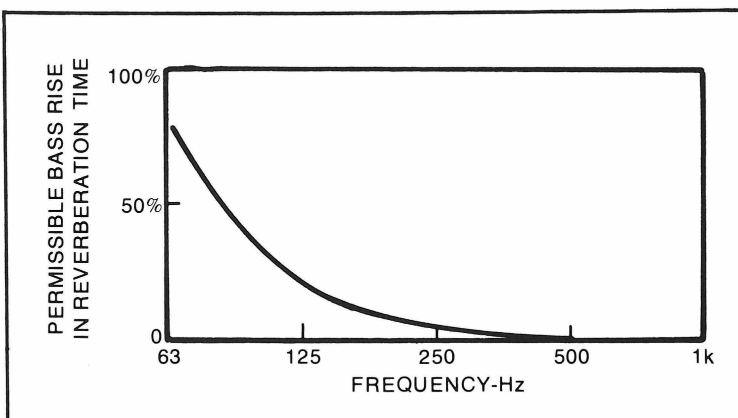

Fig. 10-5. Permissible bass rise of reverberation time for voice studios derived by subjective evaluation in controlled tests by BBC researchers. (After Spring and Randall[3].)

situation is quite different as absorbers are bulky, difficult to install, and sometimes do not behave as expected.

Researchers at the British Broadcasting Corporation observed that subjective judgements seemed to indicate a tolerance for a certain amount of bass rise of reverberation time. Investigating this in controlled tests Spring and Randall[3] found that bass rise to the extent indicated in Fig. 10-5 was tolerated by the test subjects for voice signals. Taking the 1 kHz value as reference, rises of 80% at 63 Hz and 20% at 125 Hz were found to be acceptable. These tests were made in a studio 22 × 16 × 11 ft (volume about 3900 cu ft) for which the midband reverberation time was 0.4 second (which agrees fairly well with Fig. 10-4).

Bass rise in reverberation time for music has traditionally been accepted to give "sonority" to the music in music halls. Presumably, somewhat greater bass rise than that for speech would be desirable in recording studios and listening rooms designed for classical music.

LIVING ROOM REVERBERATION TIME

The reverberation characteristic of the average living room is of interest to the high fidelity enthusiast, the broadcaster, and the recording specialist. This living room is where the high fidelity recordings are to be played. Further, the quality control monitoring room of the broadcast and recording studio must have a reverberation time not too far from that of the living room in which the final product will be heard. Generally, such rooms should be "deader" than the living room which will add its own reverberation to that of the recording or broadcast studio.

Figure 10-6 shows the average reverberation time of 50 British living rooms measured by Jackson and Leventhall[4] using octave bands of noise. The average reverberation time decreases from 0.69 second at 125 Hz to 0.4 second at 8 kHz. This is considerably higher than earlier measurements of 16 living rooms made by BBC engineers in which reverberation times between 0.35 and 0.45 were found on the average. Apparently, the living rooms measured by the BBC engineers were better furnished than those measured by Jackson and Leventhall and, presumably, would agree better with living rooms in the United States.

The 50 living rooms of the Jackson-Leventhall study were of varying sizes and shapes and degree of furnishing. The size varied from 880 to 2680 cu ft, averaging 1550 cu ft. From Fig. 10-4 we find an optimum reverberation time for speech for rooms of this size to

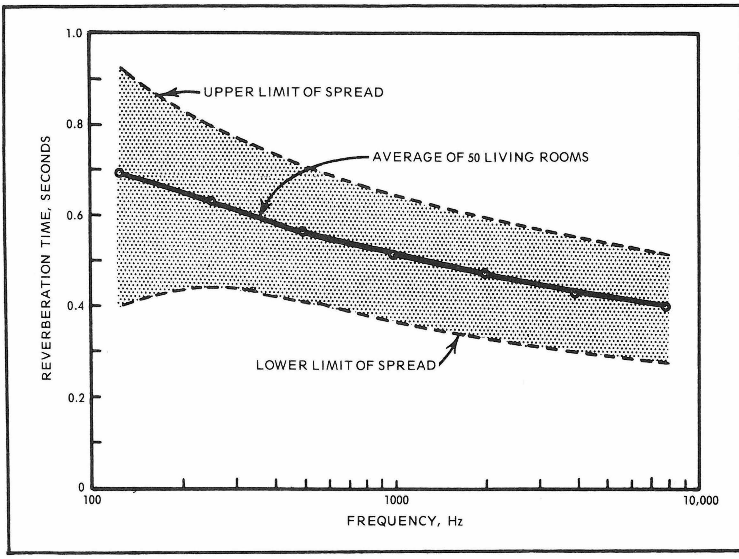

Fig. 10-6. Average reverberation time for 50 living rooms. (After Jackson and Leventhall[4])

be about 0.3 second. Only those living rooms near the lower limit approach this and in them we would expect to find much heavy carpet and overstuffed furniture. These reverberation measurements tell us little or nothing about the possible presence of colorations. The BBC engineers checked for colorations and reported serious ones in a number of the living rooms studied.

CALCULATION OF REVERBERATION TIME

It is a relatively straightforward procedure to calculate the reverberation time of an existing or proposed listening room or studio. The simplest approach is to use the reverberation equation derived by W. C. Sabine of Harvard Univeristy, a pioneer researcher in architectural acoustics:

$$T_{60} = \frac{(0.049)(V)}{(S)\,(a)} \tag{10-1}$$

in which

T_{60} = reverberation time, seconds
V = volume of room, cu ft
S = total surface area of room, sq ft
a = average sabine absorption coefficient

The quantity (0.049)(V) involves the mean free path or the average distance a sound travels between reflections. The quantity (S)(a) is the total number of absorption units (sabins) of absorption in the room. To obtain the total absorption, Sa, of the room it is necessary to combine the respective absorptions of the various materials lining the room. For example:

$$Sa = S_1a_1 + S_2a_2 + S_3a_3 \ldots\ldots \text{ etc.} \qquad \textbf{(10-2)}$$

in which

S_1 = the area of material 1
a_1 = absorption coefficient of material 1
S_2 = area of material 2
a_2 = absorption coefficient of material 2
S_3 = area of material 3
a_3 = absorption coefficient of material 3

The absorption coefficients of practically all materials vary with frequency. In a given room the areas of the different materials are different. For this reason, it is necessary to calculate Eq. 10-2 for different frequencies.

A word about the limitations of the Sabine equation, Eq. 10-1, is in order. For live rooms, statistical conditions prevail and Sabine's equation gives accurate results. What happens in very absorbent rooms? Let us take the 23.3 × 16 × 10 ft room we have used previously as an example and assume all of its surfaces are perfectly absorbent (a = 1.0). This room has a volume of 3,728 cu ft and a total surface area of 1,532 sq ft. Substituting these values in Eq. 10-1:

$$T_{60} = \frac{(0.049)\ (3{,}728)}{(1{,}532)\ (1.0)} = 0.119 \text{ second}$$

which is impossible! The mean free path 4V/S = (4) (3728)/1533 = 9.7 ft which means there would be 9.7 reflections during the reverberant decay which is impossible because perfectly absorbing walls would allow no reflections. This paradox results from assumptions upon which Eq. 10-1 is derived. As a result Eyring and others derived equations which overcame this difficulty but which are basically equivalent for average absorption coefficients of 0.25 or less.

There is also a confusion, even among the experts, concerning two ways of viewing the absorption coefficient. We would do well to

follow Young's advice[6] which avoids such problems and brings consistency to our calculations. He recommends that as long as the published absorption coefficients are obtained through use of Sabine's equation (hence called Sabine coefficients) we should use Sabine's equation, Eq. 10-1, for all calculations. It is simpler than Eyring's form and can serve our purposes so well that Eyring's equation will not even be considered further here.

Calculating reverberation time requires a bit of arithmetic, but nothing more. It is true that one's calculations may appear a bit messy from time to time because of the necessity of carrying forward work on six frequencies simultaneously, but the basic steps for each frequency are quite simple. Follow the examples step by step.

EXAMPLE 1

As our first example, let us take a completely untreated room (at the moment we do not care whether it is a listening room or studio) of the same dimensions we have used in previous examples, 23.3 × 16 × 10 ft. We further assume that the room has a concrete floor and the walls and ceiling are of frame construction with ½″ gypsum board covering. Let us forget about the door and a window for the moment as they will affect the results an insignificant amount. In Table 10-2 we shall carry forward the "as found" calculations for this room to estimate just how bad the acoustics are with no treatment added. The concrete floor area of 373 sq ft and the gypsum board area of 1,159 sq ft are entered in the table. We next refer to the appendix and enter the appropriate absorption coefficients for these two materials for each of the six frequencies. Multiplying the concrete floor area, $S = 373$ sq ft, by the coefficient $a = 0.01$ we get 3.7 sabins which is entered under Sa for 125 and 250 Hz. The absorption units (sabins) are figured for both materials for each frequency. The total number of sabins at each frequency is obtained by adding that of the concrete floor to that of the gypsum board. The reverberation time for each frequency is obtained by dividing $0.049V = 182.7$ by the Sa product for each frequency.

To obtain a graphical appreciation of the variation of reverberation time with frequency, the values are plotted in Fig. 10-7 and graph A results. We see a peak reverberation time of 3.39 seconds at 1 kHz, but considerably lower values at the extremes of 4 kHz and 125 Hz. Untreated, this room would sound quite bad. Two persons separated 10 ft would have difficulty understanding each other as the reverberation from one word covers up the next word.

Table 10-2. Room Conditions and Calculations for Example 1.

Size	23.3 × 16 × 10 ft
Treatment	None
Floor	Concrete
Walls	Gypsum board, ½″, on frame construction
Ceiling	Ditto
Volume	(23.3)(16)(10)=3,728 cu ft

Material	S sq ft	125Hz		250Hz		500Hz		1kHz		2kHz		4kHz	
		a	Sa	a	Sa	a	Sa	a	Sa	a	Sa	a	Sa
Concrete	373	0.01	3.7	0.01	3.7	0.015	5.6	0.02	7.5	0.02	7.5	0.02	7.5
Gypsum Board	1,159	0.29	336.1	0.10	115.9	0.05	58.0	0.04	46.4	0.07	81.1	0.09	104.3
Total sabins			339.8		119.6		63.6		53.9		88.6		111.8
Reverberation time Seconds			0.54		1.53		2.87		3.39		2.06		1.63

a = absorption coefficient for that material and for that frequency (See Appendix)

Sa = S times a, absorption units, sabins

$$T_{60} = \frac{(0.049)(3728)}{Sa} = \frac{182.7}{Sa}$$

Example: For 125 Hz, $T_{60} = \frac{182.7}{339.8} = 0.54$ second

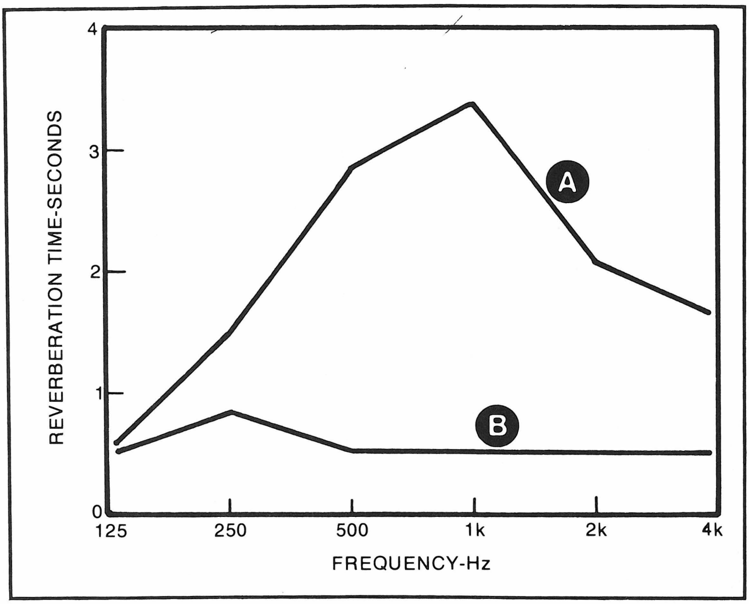

Fig. 10-7. The calculated reverberation characteristics of a 23.3 × 16 × 10 ft room: (A) the "as found", untreated condition of Example 1, (B) the treated condition of Example 2.

An important observation at this point is the dominating influence of the sound absorption by the very structure itself. The gypsum board walls and ceiling are diaphragms which are performing as resonant panel absorbers. The resonance frequency is below 125 Hz, but the 0.29 coefficient at 125 Hz indicates it may not be too far below. If the walls and ceiling were concrete like the floor the reverberation time would be very high throughout the audible frequency range.

EXAMPLE 2

Now we shall do something to control curve A of Fig. 10-7. As we study curve A it is evident that absorption is needed at mid-frequencies, very little at low frequencies, and a modest amount at high frequencies. To bring the overall reverberation time to a relatively flat condition, we need to apply a material having an absorption characteristic shaped more or less like the reverberation time curve A of Fig. 10-7. Thumbing through Chapter 9 we stop at Fig. 9-3 and note that ¾″ acoustical tile has this general shape. This is enough to convince us that it is worth trying ¾″ acoustical tile in the attempt to flatten the reverberation time curve A of Fig. 10-7.

Table 10-3. Room Conditions and Calculations for Example 2.

Size	23.3 × 16 × 10 ft
Treatment	Acoustical tile
Floor	Concrete
Walls	Gypsum board, ½″, on frame construction
Ceiling	Ditto
Volume	(23.3)(16)(10)=3,728 cu ft

Material	S sq ft	125Hz		250Hz		500Hz		1kHz		2kHz		4kHz	
		a	Sa	a	Sa	a	Sa	a	Sa	a	Sa	a	Sa
Concrete	373	0.01	3.7	0.01	3.7	0.015	5.6	0.02	7.5	0.02	7.5	0.02	7.5
Gypsum Board	1,159	0.29	336.1	0.10	115.9	0.05	58.0	0.04	46.4	0.07	81.1	0.09	104.3
Acoustical Tile	340	0.09	30.6	0.28	95.2	0.78	265.2	0.84	285.6	0.73	248.2	0.64	217.6
Total sabins			370.4		214.8		328.8		339.5		336.8		329.4
Reverberation time, Seconds			0.49		0.85		0.56		0.54		0.54		0.55

S = area of material
a = absorption coefficient for that material and for that frequency (See Appendix)
Sa = s times a, absorption units, sabins

$$T_{60} = \frac{(0.049)(3728)}{Sa} = \frac{182.7}{Sa}$$

Giving no thought at this point to how it will be distributed, how much ¾″ acoustical tile is required to correct curve A? We set up Table 10-3 to organize our calculations. Everything is identical to Table 10-2 except that the ¾″ acoustical tile is added. The coefficients for this material are picked up from the appendix which is more accurate than trying to read them off the curve of Fig. 9-3. How much tile is required? In arriving at a very rough estimate we note in Table 10-2 a total of 53.9 sabins at the peak reverberation time at 1 kHz and 339.8 sabins at 125 Hz where the reverberation time is at a reasonable 0.54 seconds. How much ¾″ acoustical tile would be required to add 286 sabins at 1 kHz? The absorption coefficient of this material is 0.84 at 1 kHz. To get 286 sabins at 1 kHz with this material would require 286/0.84 = 340 sq ft of the material, therefore we proceed to fill out Table 10-3 with 340 sq ft of ¾″ acoustical tile. Plotting these reverberation time points gives us curve B of Fig. 10-7. Obtaining reverberation time this uniform across the frequency band on the first trial is an unusual, but very satisfying achievement. Whether we make it any flatter depends on how critical will be the use of the room. Short of making measurements, the precision of our calculations is so poor that the remaining deviations in curve B of Fig. 10-7 are really not significant.

An average reverberation time of 0.55 second in a room of 3,728 cu ft volume would be quite acceptable for a music listening room according to Fig. 10-4. If it were decided that the 0.85 second point at 250 Hz should be brought down closer to 0.55 second, it would require about 100 sq ft of an absorber tuned to 250 Hz. There is no carpet on the floor. If carpet is to be used, it is an entirely new ballgame.

How should the 340 sq ft of ¾″ acoustical tile be distributed? For maximum diffusion it should be applied in irregular patches and distributed to the three axes of the room. As a first approach, the 340 sq ft could be distributed in proportion to areas on the three axes of the room as follows:

Area of N and S ends	= (2)(10)(16)	= 320 sq ft	(21%)
Area of E and W walls	= (2)(10)(23.3)	= 466 sq ft	(30%)
Area of floor/ceiling	= (2)(16)(23.3)	= 746 sq ft	(49%)

This would put (0.49)(340) = 167 sq ft on the ceiling, (0.3)(340) = 102 sq ft distributed between the east and west side walls, and (0.21)(340) = 71 sq ft distributed between the north and south end walls. One possible distribution scheme is shown in the fold-out

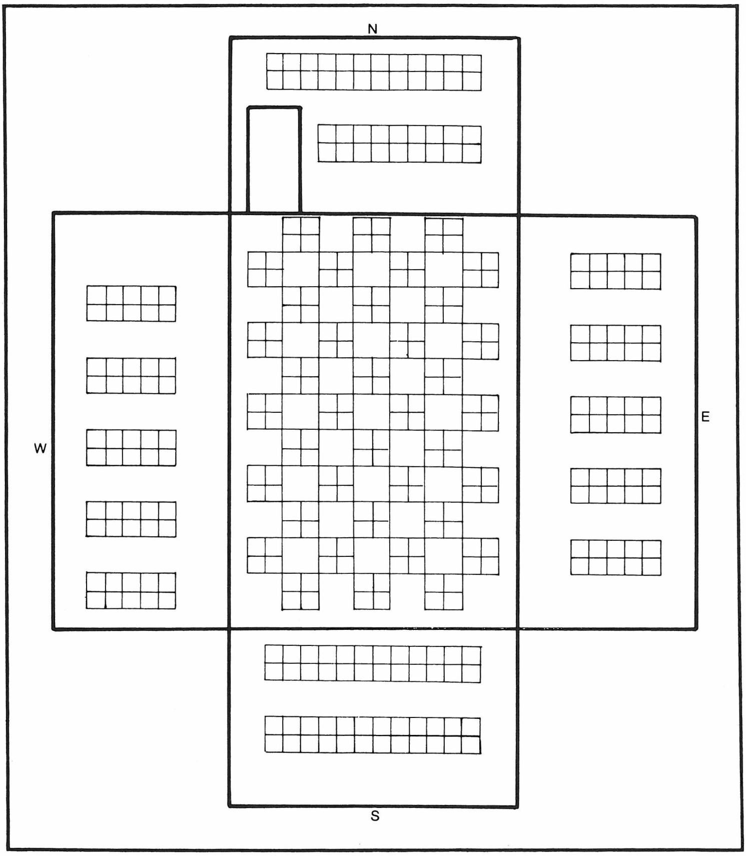

Fig. 10-8. Fold-out plan for placement of acoustical tile in 23.3 × 16 × 10 ft room as part of the exercise in Example 2.

plan of Fig. 10-8, projecting the ceiling layout to the floor. With the ceiling treated as shown, the floor must remain reflective, so the concrete is covered with vinyl tile or linoleum. Notice that the patches of tile on the north wall are opposite bare areas on the south wall and that the same alternating plan is applied to east and west walls. Whether or not you like the geometrical patterns, treating this room has shown the almost negligible absorption of the concrete and the very significant absorption by the structure and has given experience in calculating absorption and reverberation time.

The coefficient measurements are made with the standard 8 × 9 ft patch of the material under test on the reverberation room floor. Our patches are, on the average, much smaller. The increase in absorption is roughly related to increase in edge perimeter. This

effect is not exact enough to be reduced to calculations, but we can expect greater absorption from the acoustical tile by distributing them as shown in Fig. 10-8. The average reverberation time of Fig. 10-7 is about 0.63 second. The increase in perimeter would tend toward greater absorption and lower reverberation time. The 0.55 second goal might be found to be approached if measurements were made.

The astute reader may ask the question, "Should the areas of gypsum board in Table 10-3 be reduced because much of it is covered with acoustical tile?" The answer to this is that the acoustical tile is practically transparent to the low frequencies at which the gypsum board is most effective. If the gypsum board were backed by a concrete wall, it would be a very poor absorber, much like concrete alone. Its absorption on the wall framing results from diaphragmatic action. True, the acoustical tile would tend to load the diaphragm a bit, but this effect would be small. So, the answer is that the tile on the diaphragm has a very modest effect which we can safely neglect.

EXAMPLE 3: LIVING ROOM/LISTENING ROOM

In the first two examples we concentrated on one thing: how to compute reverberation time. We arbitrarily picked a 23.3 × 16 × 10 ft room with the knowledge that these were favorable ratios from Table 7-2. This time we change the dimensions to 25 × 16 × 10 ft and treat it as a living room. It is our desire to make this room a good hi-fi listening room. Of course, the room serves many other family needs and we must not upset the general decor nor allow the costs to get out of line. We shall also assume that this room is already built and we are only making alterations.

Room Proportions

First, let us see how these proportions check out as far as distribution of axial, tangential, and oblique modes are concerned. The room proportions (referring to ceiling height as 1.0) are 1.0 : 1.6 : 2.5. Checking Fig. 7-1 shows these ratios to be outside the area of acceptable ratios. What are we going to do about this? The lady of the house will probably object to shortening the room to bring it to the 1.0 : 1.6 : 2.33 favorable ratio. Can we live with it? How bad will it be and what is the nature of its faults?

Axial Modes

Proper room proportions optimize the sound diffusion in the

room considering all modes. The axial modes, however, are the most potent and can cause the most trouble. Fortunately, they are the easiest to examine in detail.

The lowest axial mode will be that associated with the greatest dimension of the room, the length, which resonates at 1130/(2)(25) = 23 Hz. The width will resonate at 1130/(2)(16) = 35 Hz and the height at 1130/(2)(10) = 56 Hz. In Table 10-4 these modes and their multiples are spread out to a frequency of about 300 Hz. Combining these in order of ascending frequency allows us to examine the spacing of the axial modes. Casting our eye down the right hand column of Table 10-4 we see modes near 70, 115, 140, 184, and 253 Hz which are separated from their neighbors by more than the 20 Hz specified in Chapter 6, but only a few Hz more. It is unlikely that these are spaced far enough to give trouble. We have a pile-up of two modal frequencies at 280 Hz, but they are well coupled to their neighbors and also very close to 300 Hz, the arbitrary point above which colorations from this source have not been observed.

Table 10-4. Axial Modes in Living Room of Example 3.

Room Dimensions:

Length . . . . 25 ft
Width . . . . 16 ft
Height . . . . 10 ft

Axial Modes - Hz

	L	W	H
f_o	23	35	56
$2f_o$	46	70	112
$3f_o$	69	105	168
$4f_o$	92	140	224
$5f_o$	115	175	280
$6f_o$	138	210	336
$7f_o$	161	245	
$8f_o$	184	280	
$9f_o$	207	315	
$10f_o$	230		
$11f_o$	253		
$12f_o$	276		
$13f_o$	299		

Combining L-W-H Axial Modes	Difference
23	12
35	11
46	10
56	13
69	19
70	22?
92	13
105	7
112	3
115	23?
138	2
140	21?
161	7
168	7
175	9
184	23?
207	3
210	14
224	14
230	15
245	8
253	23?
276	4
280	0
280	19
299	16
315	21
336	

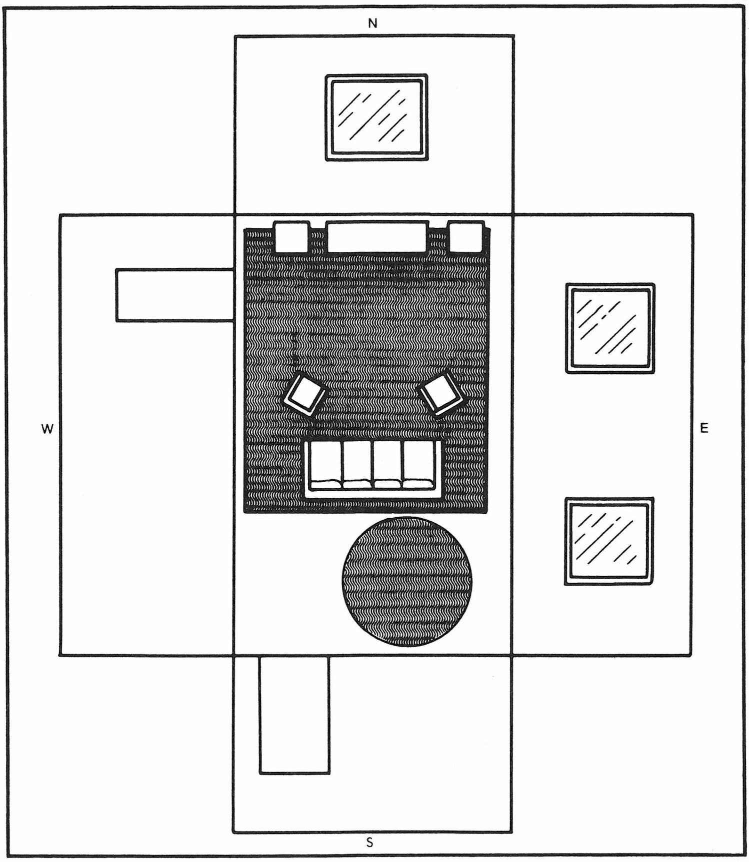

Fig. 10-9. Fold-out plan of a 25 × 16 × 10 ft living room to be considered as a home listening room in Example 3.

We can breathe a sigh of relief. Even though the room proportions are not optimized, the axial mode series seems reasonably acceptable. Further, the presence of tangential and oblique modes is helpful in filling in between the axial modes. At least it is safe to proceed until a good listening test can be made. It is interesting to note that although these particular room proportions are outside the "acceptable" area of Fig. 7-1, it so happens that they coincide exactly with the ratios recommended in earlier days by Volkmann and used in the construction of many studios.

Room Furnishings

As this room is the living room of a residence, we shall specify a wood floor and walls and ceiling of plaster on lath. The fold-out

sketch of the room, Fig. 10-9, includes only those furnishings having a significant acoustical absorption.

Actually, every small table, straight-backed chair, and picture on the wall contribute to diffusion of sound in the room, but only the softer items such as sofas, upholstered chairs, rugs, and drapes absorb much sound.

We can certainly hope that the hi-fi enthusiast in the family would have something to say about the arrangement of the furniture. He would probably prefer having his loudspeakers aimed down the long dimension of the room as shown in Fig. 10-9 and Harry Olson tells us that they should be spaced about 0.7 of the room width, about 11 ft in this case. Placing the sofa about a room width away is within the good listening area. A couple of upholstered chairs placed as shown would help increase the seating capacity, but would provide somewhat less desirable positions. A 14 × 16 ft rug covers the north end of the room and a 7½ ft circular rug in the south end is a concession to a games area or a place to curl up with a good book.

Reverberation Calculation

Now comes the fun. What do we have to do with this living room to come out with an optimum reverberation time of about 0.5 second throughout the audible band? It is instructive to treat the problem in several stages, watching the effect on reverberation time as new things are added. Table 10-5 shows the calculations for the six standard frequencies. Stage 1 includes the following:

Table 10-5. Room Conditions and Calculations for Example 3.

Size	25 × 16 × 10 ft	Ceiling	plaster on lath
Floor	wood	Volume	(25)(16)(10) = 4,000 cu ft
Walls	plaster on lath		

Material	S sq ft	125Hz a	125Hz Sa	250Hz a	250Hz Sa	500Hz a	500Hz Sa	1kHz a	1kHz Sa	2kHz a	2kHz Sa	4kHz a	4kHz Sa
Glass	65	0.35	22.8	0.25	16.2	0.18	11.7	0.12	7.8	0.07	4.6	0.04	2.6
Plaster	708	0.14	99.1	0.10	70.8	0.06	42.5	0.05	35.4	0.04	28.3	0.03	21.2
Wood	132	0.15	19.8	0.11	14.5	0.10	13.2	0.07	9.2	0.06	7.9	0.07	9.2
Carpet	268	0.02	5.4	0.06	16.1	0.14	37.5	0.37	99.0	0.60	161.0	0.65	174.0
Sofa			12.0		19.0		30.0		38.0		48.0		45.0
Chairs (2)			4.8		7.6		12.0		15.2		19.2		18.0
Stage 1 total sabins			163.9		144.3		146.9		204.6		269.0		270.0
Drapes	192	0.07	13.4	0.31	59.5	0.49	94.0	0.75	144.0	0.70	134.0	0.60	115.0
Stage 1 and 2 total sabins			177.3		203.8		240.9		348.5		403.0		385.0
Paneling	700	0.55	384.0	0.35	245.0	0.11	77.0	0.08	56.0	0.07	49.0	0.06	42.0
Stage 1, 2 & 3 total sabins			561.3		448.8		317.9		404.5		452.0		427.0
Reverberation Time													
Stage 1			1.20		1.36		1.33		0.96		0.73		0.73
Stage 2			1.11		0.96		0.81		0.56		0.49		0.51
Stage 3			0.35		0.44		0.62		0.48		0.43		0.46

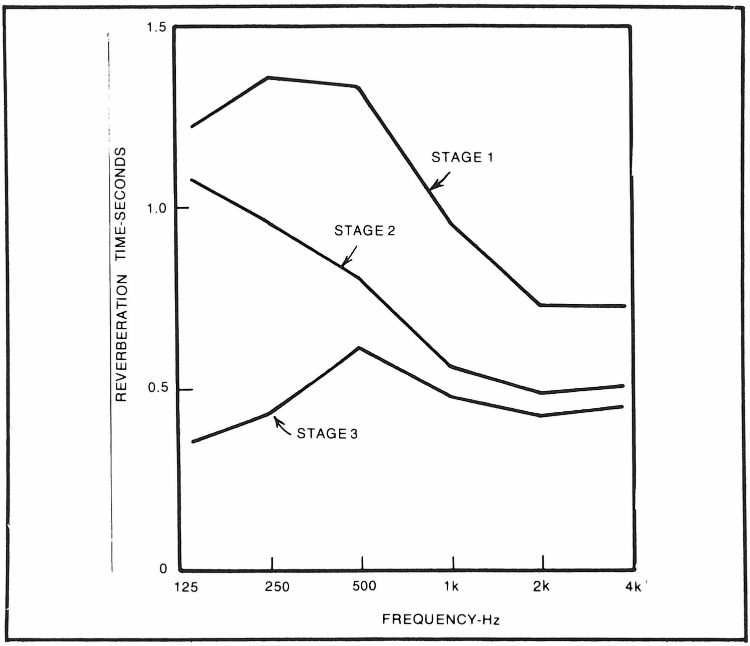

Fig. 10-10. Stage-by-stage changes in the calculated treatment of the 25 × 16 × 10 ft living room of Fig. 10-9 (Example 3): (1) including rugs, sofas, chairs, (2) adding drapes, (3) adding wood paneling.

Glass. Window panes vibrate as diaphragms and have good low frequency absorption.

Plaster. Large area and significant contribution to low frequency absorption.

Wood. Floor only, the areas of the doors is small enough to neglect their contribution.

Rugs. Heavy with no padding. Use carpet coefficients from the appendix.

Sofa. Now here is a tough one. There are no absorption coefficients to fit the thousands of kinds of sofas extant. From some out of date literature not listed are given absorption units (not coefficients) for padded theater seats. A rough guess has been made that the sofa has ten times the area and hence ten times the absorption of a padded theater seat. Another educated guess approach would estimate the equivalent surface area of the sofa and imagine it to be some very absorbent material which is listed in the appendix such as heavy carpet. Remember that the ends and back also absorb and even the underside if the sofa is up on legs.

Chairs. Two upholstered chairs are each estimated to have twice the absorption of a padded theater seat.

We then total the absorption units (sabins) for the above listed items in Stage 1 and figure the reverberation time for each frequency by dividing $(0.049)(4000) = 196$ by the total sabins for each frequency. The result is plotted in Fig. 10-10. At the low frequency end the reverberation time is about 1.2 seconds and it is about 0.73 seconds in the highs. Of course this excessive reverberation time must be brought down.

In Stage 2 we add drapes with a height of 8 ft. The north window drapes are 9 ft wide and the east window drapes 7.5 ft wide giving a total area of 192 sq ft of medium velour draped to half area (coefficients in the appendix). This brings the reverberation time down a significant amount, but not nearly enough. The introduction of rugs, drapes, and some upholstered furniture leaves us short of our reverberation goal of about 0.5 second.

In Stage 3 we add 700 sq ft of ⅛″ paneling furred out from the plaster wall 3 inches with 1 inch of glass fiber absorbent in the cavity. This panel resonates at about 150 Hz (Fig. 9-16). If we cover all the walls with paneling, of course we should go back and subtract the absorption of the plaster on the walls which has been covered. However, the plaster on lath is also a diaphragm as indicated by the rise in absorption at the lower frequencies. This diaphragm action would persist, even though a second diaphragm (the paneling) is placed on top of it because of the penetrating power of low frequency energy. For this reason, our best course of action is to leave the plaster absorption in the columns. The reverberation graph for Stage 3 is shown in Fig. 10-10. Not too bad! We have a slight peak at 500 Hz, but otherwise this graph is about as good as can be expected. Measurements of reverberation time would certainly be the next stage to provide the basis for trimming with real meaning.

Following the above procedure the reader can trace the various steps, making whatever changes are necessary to achieve the desired results.

There is a bit of mystery here. Why did we have to add paneling to the walls to bring the low frequency reverberation time down? The average reverberation time of 50 British living rooms shown in Fig. 10-6 shows something of the tendency to rise in the lows. We have no data on the individual houses but probably many are apartments of concrete or block construction lacking the diaphragmatic action of frame construction. The area of wood floor used in Stage 1

is that not covered by rugs, while in reality the floor as a diaphragm acts that way whether there is a rug on it or not.

Wood paneling has historically been associated with music rooms of Europe famous for their acoustics. The vibration of these panels seems to envelop the listener in the music. For this reason wood paneling is a natural and a logical material for achieving low frequency absorption in music rooms for performing or listening. A thin veneer of an exotic wood can make such panels very beautiful.

The length of this living room is about 1 ft-8 inches longer than the preferred length of Table 7-2. If the lady of the house would go along with the idea, a false wall could be installed with, say, 2 × 4 framing with gypsum board surface covering and the loudspeakers could be mounted flush in this wall. Recessed shelves for books, records, or equipment could also be made a part of this wall. Lining the cavity behind with a generous area of inexpensive glass fiber building insulation would be wise to reduce the tendency for resonances to build up in the cavity. The loudspeakers should be mounted at about seated ear level.

We have said nothing about protecting our listening room from disturbing noises originating outside the room. In fact, if you like a big sound, the rest of the house may need to be protected from the music room! It is possible that staggered stud walls and solid core doors would be advisable in some cases.

Something should be said about the acoustical coupling of two spaces. Opening a door of normal size would be rather a small coupling between our listening room and the space into which the door opens. With a larger opening, however, the reverberation characteristics of the coupled space could react on those of the listening room in a rather complicated way. The simplest solution, if such trouble develops, is to close the door!

We left our paneled walls flat and we know that some irregularities are needed to avoid flutter echo. Oil paintings are excellent for this and for a bit of low frequency absorption of their own (stretched diaphragms with air space behind). Other knickknacks, gimcracks, and bric-a-brac will help also in diffusion. The wall opposite the loudspeakers may need some special attention. How about hanging a nice tapestry?

EXAMPLE 4

To stretch our minds a bit in the calculation of reverberation time, let us take the same 25 × 16 × 10 ft room and make a

recording studio out of it. Let us assume that the primary interest is in conventional music produced by small vocal and instrumental ensembles. Going back to Fig. 10-4 we would select for such use a reverberation time objective at the upper edge of the shaded area to favor music. For a volume of 4,000 cu ft this would be near 0.6 second.

The proportions are not optimum for diffusion of sound according to Table 7-2. The C set of ratios would call for a room length of 23.3 ft instead of the existing 25 ft. We could shorten the room but, as we saw in Example 3, leaving it 25 ft long created no serious axial mode problems, so we shall not shorten it.

In Chapter 9 we considered many different materials and structures capable of giving the low frequency absorption we know is needed in a small studio. Because it is primarily a music studio it is natural for us to think of paneling and because we are seeking good diffusion and brilliance for a recording studio, our thoughts go toward polycylindrical diffusers. It has been mentioned that polys are not very popular today, but does anything as functional as polys really ever go out of style? In any event, we choose to go the poly route in this studio.

Where does one start? First it is necessary to do some general thinking. Thoughts like this are in order:

(a) Let's see, a reverberation time of 0.6 second is desired. What absorption is required to get it? For a studio of 4,000 cu ft volume, this will require how many sabins? From Eq. 10-1 we compute this to be 327 sabins.

(b) Now, we will need 327 sabins at each of the six frequency points. Polys can provide a large part of the low frequency absorption and some needed diffusion as well. How many polys are needed to do the job?

Looking at the poly absorption graphs of Fig. 9-18 we see in the 100-300 Hz region the absorption coefficient is 0.3 to 0.4; let's take an average of 0.35 as we will use polys of different sizes. To estimate the area of polys required we use the relationship:

$$\text{Absorption units (sabins)} = \text{(area)(absorption coeff.)}$$

$$\text{sabins} = Sa$$

$$S = \frac{327}{0.35} = 934 \text{ sq ft.}$$

Not all the low frequency absorption will be from polys. The ceiling alone has 400 sq ft, so it would appear that we have enough surface to accommodate 934 sq ft of polys.

(c) For high frequency absorption let us consider common acoustical tile again such as used in Example 2.
(d) Ideally we should have some of each acoustical material applied to every surface of the studio. Practically, we can only approach this. For one thing, the only practical acoustical material for the floor is carpet and having the floor covered with carpet gives more absorption in the highs than we can stand.

A floor area of 400 sq ft and carpet absorption coefficient of about 0.7 gives 280 sabins in the highs for the carpet alone and the polys have coefficients of 0.2 in the highs also. Even if we used only 500 sq ft of polys, they would contribute another 100 sabins. Carpet plus polys would give $280 + 100 = 380$ sabins and we only want 327.

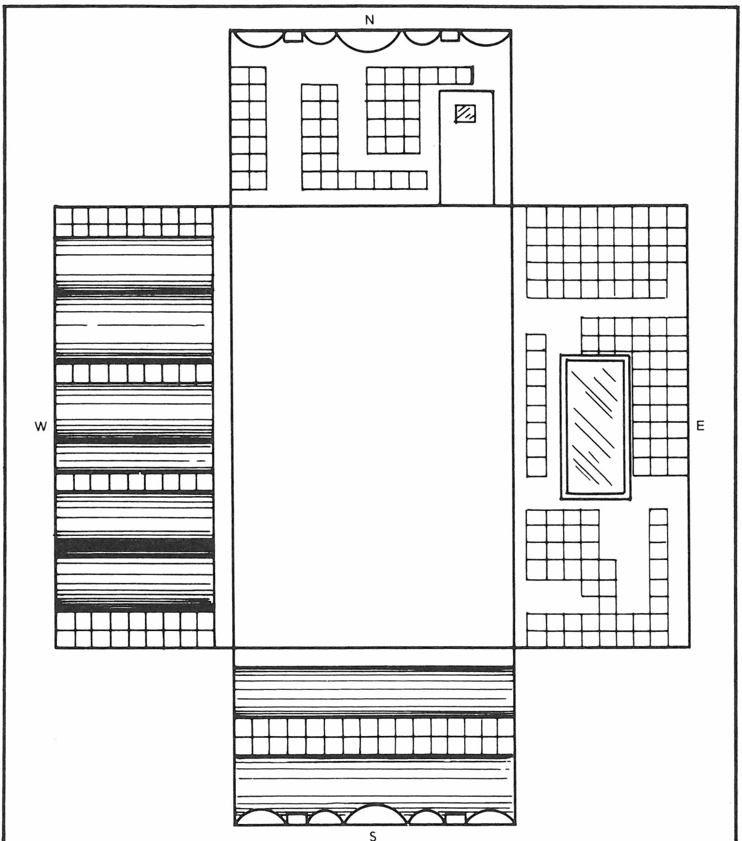

Fig. 10-11. Fold-out plan of 25 × 16 × 10 ft music studio of Example 4. Note that the axes of the polys on the walls and ceiling are all mutually perpendicular.

So, carpet is out. We'll settle for a throw rug directly under the microphone.

(e) We've theorized and nibbled around the edges about as much as we can, now we must dive into some preliminary, exploratory calculations. Basically, it is a cut-and-try process. The more experience one has, the fewer the exploratory calculations required. Taking the poly situation first, we know we should use polys of different chord lengths; let's pick three of the four shown in Fig. 9-18, poly A, poly B, and poly D. With a foldout sketch of the studio drawn to scale, you are ready to sketch possible configurations. In fact, it is handy to have a dozen photocopies of the fold-out sketch to scribble on. Figure 10-11 is about the third or fourth such sketch and there were calculations for each.

On the north and south walls the cross section of the ceiling polys are visible. A big poly A is in the center, flanked with a poly D on each side. Between the D polys and the outside B polys is space for fluorescent or other lighting fixtures running the length of the room. This well-polyed ceiling opposes the hard, reflective floor covered with vinyl tile and should completely care for flutter in the vertical direction.

On the west wall two A polys, three B polys, and one D poly are arranged vertically which places them perpendicular to the ceiling polys as they should be. Four vertical tiers of acoustical tile are worked into this west wall, two of them at the corners where absorbers are especially effective. This breaks up the west wall so that we would expect no flutter between it and the east wall.

The south wall has a single A and a single B poly arranged horizontally so that the axis of each set of polys is perpendicular to the other two sets. Two horizonal tiers of acoustical tile are placed between 4 and 6 ft from the floor (which is head height for a standing person). Good high frequency absorbers are placed at this height on all walls except the one to which a performer would turn his back, the west wall.

(f) With the polys as selected in Fig. 10-11 and using the absorption coefficients plotted in Fig. 9-18 and tabulated in the appendix, the absorption at the six standard frequencies was calculated (Table 10-6). The absorption attributed to the polys is the lower area of Fig. 10-12.

The ¾″ acoustical tile offers an absorption coefficient of 0.73 in the highs and, following the same procedure as with the polys in (b) we find that we need $200/0.73 = 274$ sq ft of acoustical tile. The upper part of Fig. 10-12 is plotted from the calculations of Table

Table 10-6. Room Conditions and Calculations for Example 4.

Size 25 × 16 × 10 ft
Floor vinyl tile
Walls plaster
Volume (25) (16) (10) = 4,000 cu ft

	Material	S sq ft	125 Hz		250 Hz		500 Hz		1 kHz		2 kHz		4 kHz	
			a	Sa	a	Sa	a	Sa	a	Sa	a	Sa	a	Sa
EMPTY	Poly A	232	0.41	95.1	0.40	92.8	0.33	76.6	0.25	58.0	0.20	46.4	0.22	51.0
	Poly B	271	0.37	100.3	0.35	94.9	0.32	86.7	0.28	75.9	0.22	59.6	0.22	59.6
	Poly D	114	0.25	28.5	0.30	34.5	0.33	37.6	0.22	25.1	0.20	22.8	0.21	23.9
	Total			223.9		222.2		200.9		159.0		128.8		134.5
FILLED	Poly A	232	0.45	104.4	0.57	132.2	0.38	88.2	0.25	58.0	0.20	46.4	0.22	51.0
	Poly B	271	0.43	116.5	0.55	149.1	0.41	111.1	0.28	75.9	0.22	59.6	0.22	59.6
	Poly D	114	0.30	34.2	0.42	47.9	0.35	39.9	0.23	26.2	0.19	21.7	0.20	22.8
	Total			255.1		329.2		239.2		160.1		127.7		133.4
	Acoustical Tile, ¾"	274	0.09	24.7	0.28	76.7	0.78	213.7	0.84	230.2	0.73	200.0	0.64	175.4
Total sabins EMPTY				248.6		298.9		414.6		389.2		328.8		309.9
Total sabins FILLED				279.8		405.9		452.9		390.3		327.7		308.8
Reverberation Time EMPTY				0.79		0.66		0.47		0.50		0.60		0.63
Reverteration Time FILLED				0.70		0.48		0.43		0.50		0.60		0.63

10-6. This area of tile was then distributed around the studio in patches as shown in Fig. 10-11. There is nothing magical in this arrangement, only the knowledge that random patches contribute to diffusion of sound.

(g) Figure 10-12 shows a very slight deficiency of absorption in the

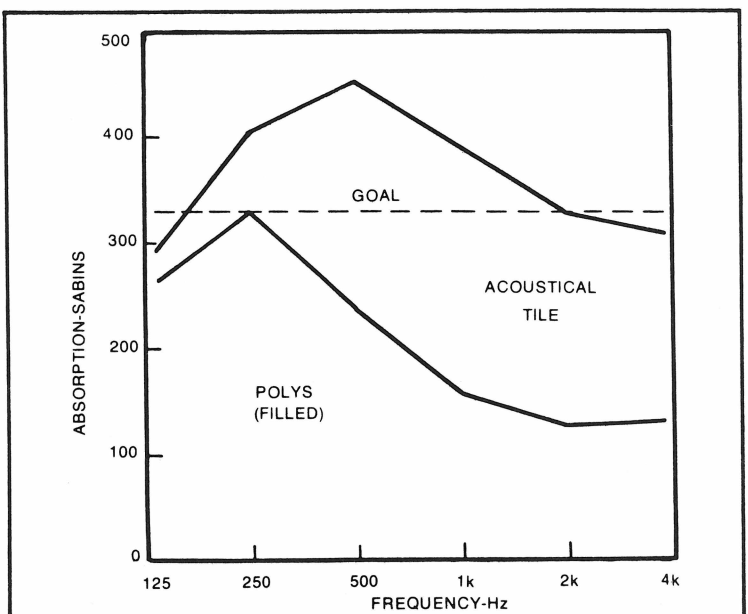

Fig. 10-12. The relative contributions of the polys and the acoustical tile to the overall absorption in the music studio of Fig. 10-11, Example 4.

lows and considerable extra sabins midband. Figure 9-18 shows that stuffing the polys full of glass fiber increases absorption in the lows. Will this help us? Table 10-6 carries forward calculations for both empty and filled polys and the effect on the reverberation time graph is shown in Fig. 10-13.

We must always be aware of the lack of precision in the calculations such as just presented. The curves appear to be precise and graceful. The calculations have been carried to three and four significant figures (except that we fall back to two significant figures for reverberation time). One reverberation time graph in Fig. 10-13 is about as good as the other when all the uncertainties are considered. It takes measurements to know how to trim up such a design. Therefore, there is no point to straining further. Sounds recorded in this studio will almost certainly be very acceptable, but we should always be alert to the possibility of colorations due to some persistent axial mode.

THE "SOFT" STUDIO

Many studios today have highly absorbent treatments which seem to negate all we have been learning about optimum reverberation time, sound diffusion, etc. These "dead" rooms are used for very special types of recording, primarily multitrack recording of rock music groups. The various instruments and vocalists are ar-

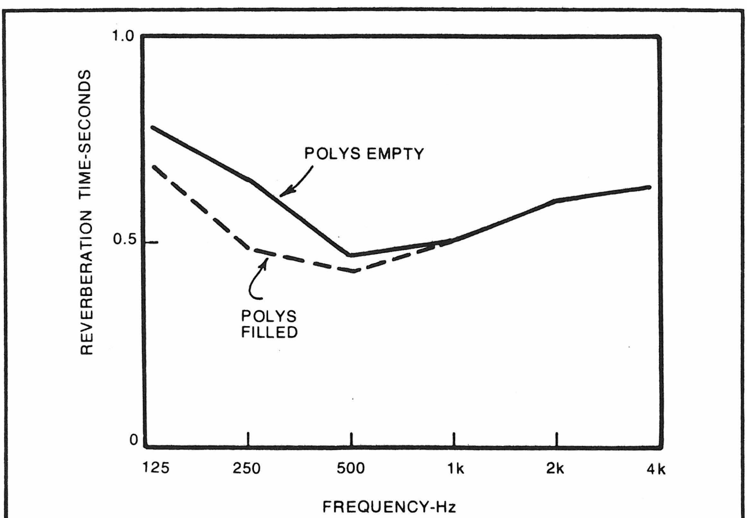

Fig. 10-13. The resulting calculated reverberation characteristics of the 25 × 16 × 10 ft music studio of Fig. 10-11, Example 4.

ranged for maximum track separation by distance and the use of flats. Each is recorded on a separate track for later combination. This gives the producer complete freedom to achieve any balance in the mix he wishes as well as the opportunity of introducing any desired amount of special equalization, stereo effect, filter effects, or artificial reverberation.

In such studios much of the walls and ceiling are covered with 4 inches of glass fiber, much as the conventional treatment of motion picture sound stages. Surrounding each performer with plywood flats provides some sound return which enables the performer to hear his or her own instrument or voice. Without such "local" acoustics there is a tendency for a performer to feel he or she is not producing enough sound, resulting in a strained effect to correct the situation.

TWO ROOMS COUPLED ELECTROACOUSTICALLY

What is the overall reverberant effect when sound picked up from a studio having one reverberation time is reproduced in a listening room having a different reverberation time? Does the listening room reverberation affect what is heard? The answer is definitely yes. This problem has been analyzed mathematically by Mankovsky.[7] In brief, the sound in the listening room is affected by the reverberation of both studio and listening room as follows:

(a) The combined reverberation time is greater than either alone.
(b) The combined reverberation time is nearer the longer reverberation time of the two rooms.
(c) The combined decay departs somewhat from the straight line of Fig. 10-2.
(d) If one room has a very short reverberation time, the combined reverberation time will be very close to the longer one.
(e) If the reverberation time of each of the two rooms alone is the same, the combined reverberation time is 20.8% longer than one of them.
(f) The character and quality of the sound field transmitted by a stereo system comforms more closely to the mathematical assumptions of the above than does a monaural system.
(g) Items (a) to (e) can be applied to the case of a studio linked to an echo chamber as well as a studio linked to a listening room.

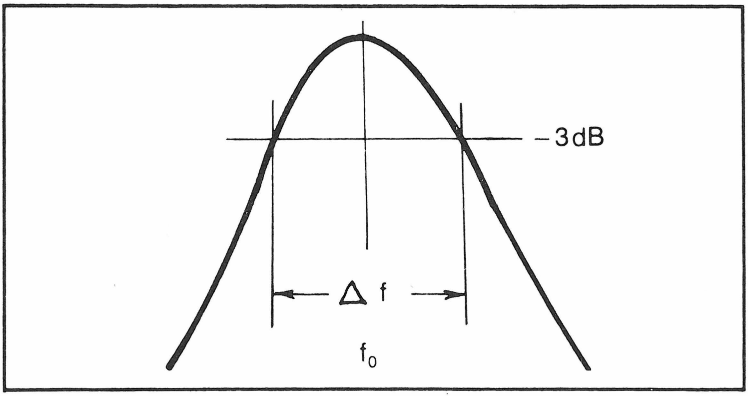

Fig. 10-14. Once the tuning curve of a Helmholtz type resonant absorber has been determined, its Q-factor may be found from the expression, $f_0/\Delta f$. The "reverberation time" of such absorbers is very short for Q's normally encountered.

REVERBERATION TIME OF HELMHOLTZ RESONATORS

Some concern has been expressed about the possibility of acoustically resonant devices, such as Helmholtz absorbers, "ringing" with a "reverberation time" of their very own adding coloration to the voice and music signals. It is true that any resonant system, electronic or acoustical, has a certain time constant associated with it. The Q-factor (quality factor) describes the sharpness of tuning of the Helmholtz resonator as shown in Fig. 10-14. Once the tuning curve has been obtained experimentally, the width of the tuning curve at the -3 dB points gives Δf. The Q of the system is then $Q = f_0/\Delta f$, where f_0 is the frequency to which the system is tuned. Measurements on a number of perforated and slat Helmholtz absorbers gave Q's around 1 or 2 but some as high as 5. Table 10-7 shows how the decay rate of resonant absorbers of several Q's comes out expressed in terms of reverberation time.

With resonant absorber Q's of 100, real problems would be encountered in a room having a reverberation time, say, of 0.5 second as the absorbers tailed off sound for several seconds. However, Helmholtz absorbers with such Q's would be very special devices, made of ceramic, perhaps. Absorbers made of wood with

Table 10-7. Sound Decay in Resonant Absorbers.

Q	f_0	"Reverberation Time"
100	100 Hz	2.2 seconds
5	100	0.11
1	100	0.022

glass fiber to broaden the absorption curve have Q's so low that their sound dies away much faster than the studio or listening room itself.

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11

Acoustical Design of Studios

Tracing each step in the acoustical design of a typical suite of general purpose studios and their associated control rooms is the basic plan of this chapter and the one to follow. This should give an appreciation of the type of problems encountered and the inevitable compromises involved.

The general steps in any studio design include: (1) working with the client to determine space requirements and location of facility, (2) evaluation of environmental noise and air conditioning noise exposure at the selected site, (3) conferences with the client on layout, including number of studios, control rooms, and auxiliary support space, (4) selection of volume and shape of rooms to assure satisfactory modal frequency distribution, and (5) acoustical treatment of the spaces.

The examples selected for study are taken directly from the actual design of the new studio facilities for the Pacific Broadcasting Association (PBA) in Tokyo, Japan. These studios are used for recording of speech and musical programs for radio broadcasting. They are located on the seventh floor of a newly constructed high rise building in a heavily congested area of Tokyo. The acoustical design was by the author with excellent support from Mr. Steve Tygert, director, and the PBA staff in accomplishing the design goals in the face of availability of materials and meeting local codes. The fire regulations in Tokyo are among the most strict of comparable cities in the world, no doubt in response to a number of devas-

tating fires in recent years. The selection of a foreign suite of studios may make comparison with domestic studios a bit difficult at times, yet, on the other hand, it will provide a wide range of examples of creative engineering to meet unusual problems. In other words, the emphasis is placed more on the dynamics of acoustical design rather than on specific, cut-and-dried solutions.

GENERAL LAYOUT

The floor plan finally selected is shown in Fig. 11-1. The two small speech studios, A and B, are of minimum volume (1,424 cu ft) for space economy. The Studio C, to be used primarily for music recording, is considerably larger (5,154 cu ft) to accommodate the size of musical groups contemplated. Also as an economy, to Control Room B was assigned the control function of both Studio B and Studio C. This was judged to be operationally feasible because of the less frequent use of Studio C. The auxiliary room is more a general work room to take the pressure off the control rooms in such functions as tape dubbing, editing, etc.

The heating, ventilating, and air conditioning (HVAC) room was located in such close proximity to the sound sensitive area as a necessity, and not by design. Only extreme measures give this location a chance of success. This HVAC room is set off by double walls on two sides. The machinery rests on isolation mounts on a floating concrete floor. The short runs of air supply and return ducting are made workable only by specially designed plenum sound traps in the ducts (designed by a local contractor).

A basic problem for control rooms and Studios A and B was the low ceiling height. From concrete structural floor surface to the bottom of the concrete structural floor above (maximum clearance) is only 11 ft and the usable height is greatly reduced by massive structural beams in the most inconvenient places. Studio C is located in a separate section of the building in which it was possible to encroach on the floor above. This yielded the required height to achieve optimum proportions.

STUDIO-A DESIGN

It is difficult to imagine the magnitude of noises carried in the very structural members of a building to every part of the building. In traveling through reinforced concrete, sound is attenuated only a few dB per 100 ft. Motors and pumps starting and stopping, doors slamming, plumbing noises, footsteps, scraping of chairs, and business machines are but a few of the noise sources in the typical high

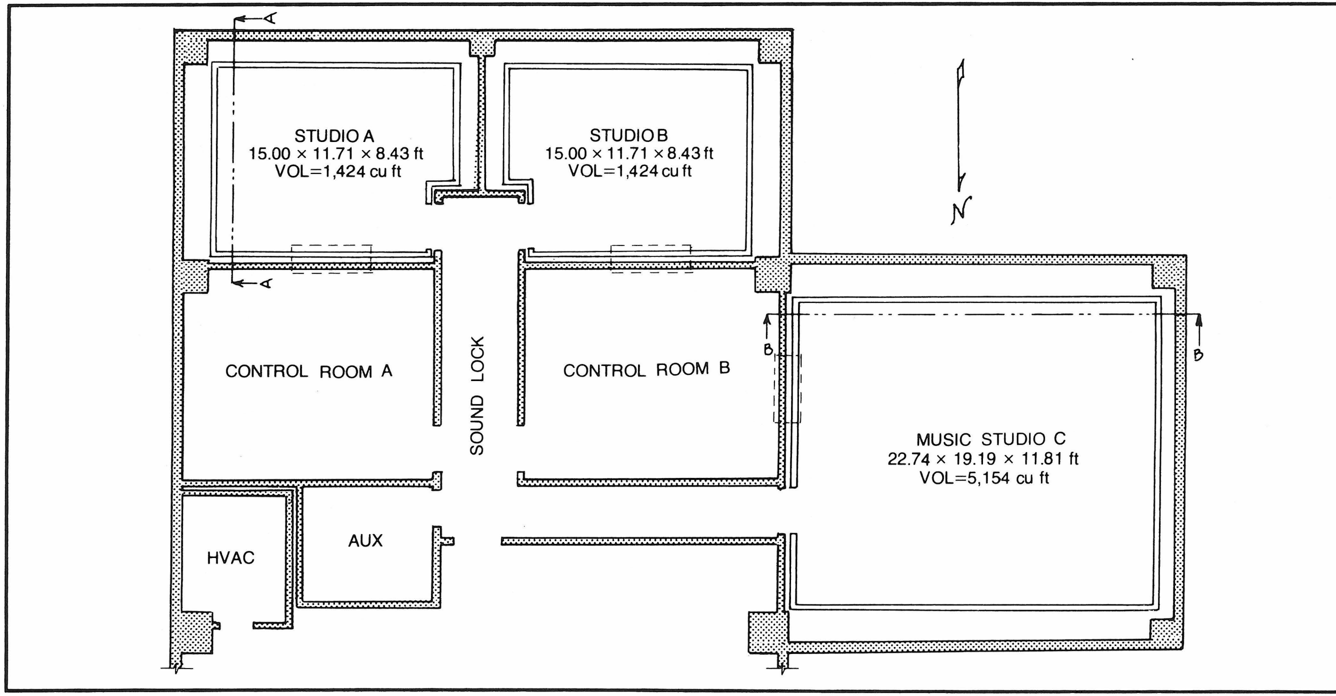


Fig. 11-1. Floor plan of studio suite of the Pacific Broadcasting Association (PBA) in Tokyo, Japan. Studios A and B are speech studios, Studio C is for music recording.

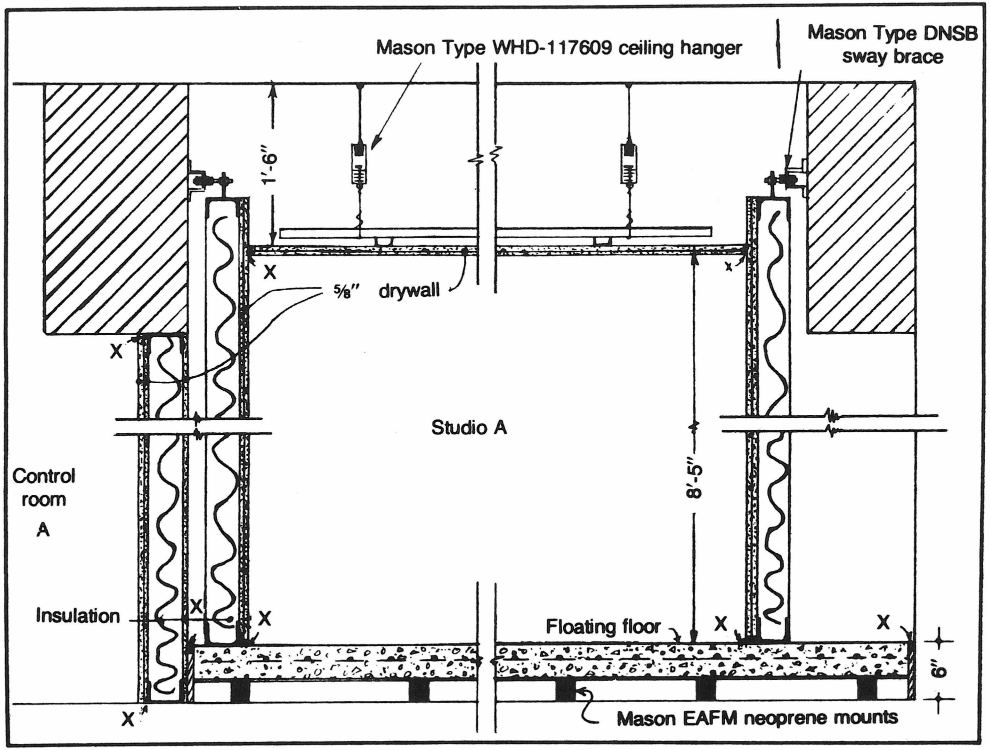


Fig. 11-2. Section A-A of Fig. 11-1 of PBA Studio A walls and floor. The floating floor of reinforced concrete supports the four walls. The ceiling is resiliently supported from the building structure.

rise situation. Such structural vibrations, carried throughout the building with high efficiency, set concrete walls to vibrating. Their large area results in efficient radiation of the vibrational energy as airborne sound.

Because of high noise levels existing in typical concrete structures, the "room-within-a-room" technique was recommended for all studios. Floating concrete floors, walls supported on the floating slab, and ceilings resiliently supported from the structure were specified as shown in Fig. 11-2.

FLOATING FLOORS

It was determined by the building architect that the structural floor could bear the additional load of the floating floor. Numerous preparatory steps are necessary before the floating floor is poured. A special board of compressed glass fiber 1″ thick is installed at the perimeter of the floating floor to isolate the slab from the structure at the edges. Compressed glass fiber cubes or molded neoprene or rubber "hockey pucks" are distributed so that the compression of the mounts and the loading are carefully matched. Sheets of marine plywood of ½″ thickness are then laid on the resilient mounts and their edges fastened together with metal strips and screws. The plywood is then covered with a heavy plastic sheet, overlapping a foot or more at the joints and running up over the perimeter board. Welded screen reinforcing mesh is put in place and blocked up to be at about the center of the finished slab. The floating floor is then poured. After the concrete has set the plastic sheet is cut and the periphery sealed with a non-hardening sealant. This sealing operation is aided if a ½″ strip of wood is first set on top of the perimeter board so that, when removed after pouring, there results a neat ½″ slot to be filled with sealant.

WALLS

Two layers of ⅝″ gypsum board (drywall) is the standard wall and ceiling surface in Fig. 11-2. The walls are supported on the conventional metal studs which, in turn, rest on the floating floor. Between Studio A and the Control Room A greater isolation is required, justifying the double wall construction. As the wall on the control room side is installed before Studio A floor is poured, drywall is mounted on both sides of the studs, double toward the control room, single toward Studio A. Normal building thermal insulation of 4″ thickness is mounted in all walls. The top of each

wall section is secured by resilient sway braces connecting to nearby structural members.

CEILING

Studio A ceiling is a conventional suspended ceiling of double ⅝″ gypsum board, but with two unconventional features. The metal frame to which the drywall is fastened is supported from the structure with wires in the usual way, but an isolating ceiling hanger is mounted in each support wire. The second feature is the acoustical sealant around the periphery of the ceiling.

ROOM PROPORTIONS

It may not be apparent from Fig. 11-1, but it is obvious in Fig. 11-2 that to achieve sufficient height in Studio A the walls must be pulled in until they are not under the large beams. A considerable amount of exploration was required to get both Studio A and Studio B in the overall width of the building and with favorable room proportions within the constraints of the beams. Using the axial mode method of study, as shown in Table 11-1, the dimensions of

Table 11-1. Axial Modes in Studio A.

Room Dimensions: Length, 15.00 ft; Width, 11.71 ft; Height, 8.43 ft

Axial Mode Resonances				Arranged in Ascending Order	Axial Mode Spacing Hz
	L	W	H		
f_1	37.7	48.2	67.0	37.7	10.5
f_2	75.3	96.5	134.0	48.2	18.8
f_3	113.0	144.7	201.1	67.0	8.3
f_4	150.7	193.0	268.1	75.3	21.2
f_5	188.3	241.2	335.1	96.5	16.5
f_6	226.0	289.5		113.0	21.0
f_7	263.7	337.7		134.0	10.7
f_8	301.3			144.7	6.0
				150.7	37.6
				188.3	4.7
				193.0	8.1
VOLUME = 1,481 cu ft				201.1	24.9
				226.0	15.2
MEAN SEPARATION OF				241.2	22.5
AXIAL MODES = 15.51 Hz				263.7	4.4
STD. DEVIATION = 8.8 Hz				268.1	21.4
				289.5	11.8
				301.3	

15.00 × 11.71 × 8.43 ft were established for both Studio A and Studio B. Correcting for the doorway jog, a net volume of 1,424 cu ft resulted for each room.

The separation of adjoining axial modal frequencies, shown in the right hand column of Table 11-1, shows reasonably good axial mode distribution (minimum 6 Hz, maximum 21.1 Hz) out to 150 Hz. In Fig. 6-13 we noted that the frequencies at which most colorations occur are between about 100 and 180 Hz. It would appear we are quite safe to 150 Hz, but the 193 and 201 Hz modes, being well isolated by the 37.6 Hz separation on one side and 24.9 Hz on the other, might just possibly yield a coloration. Being forewarned, careful listening will tell whether such a concern is justified. Not all potential colorations really develop. It is interesting to note that the proportions of Studio A (1.78; 1.39; 1.00) fall squarely in the Bolt area of Fig. 7-1, but do not coincide with any of Sepmeyer's ratios of Table 7-2.

ACOUSTICAL TREATMENT OF STUDIO A

Many of the types of acoustical treatment used in the United States do not pass Tokyo fire regulations. Other familiar materials are either not available in Japan or are too expensive. Fortunately, the Japanese have a well-developed acoustical materials industry. It merely takes a bit of searching to learn the characteristics of their products. Only the final selections will be described.

A 24″×24″×¼″ Aslux panel offered by the Nichi Company is acceptable from a fire standpoint. It comes unperforated or in standard perforations of 6.9% and 28%. The availability of these panels suggested the possibility of covering the walls with them mounted on Nagasawa NS steel channels 4″ × 1.75″ × 1/32″ set on edge. These are light enough to be cemented to the drywall surface. The north and west walls and the east wall to the entrance jog are covered with a grid of these lightweight steel channels 24″ × 24″ on centers. A 3 × 3 grid is also cemented to the ceiling as shown in Fig. 11-3. Aslux panels unperforated or with various perforations serve as covers for each square of the grid.

Studio floors are covered with a lightweight felted type of carpet similar to our indoor-outdoor carpet. Carpet absorption is a dominant factor at higher audio frequencies. The walls and ceiling of double ⅝″ gypsum board provide diaphragmatic absorption at the lower audio frequencies. The 24″ × 24″ wall and ceiling modules are then designed to smooth out the absorption curve at a level that

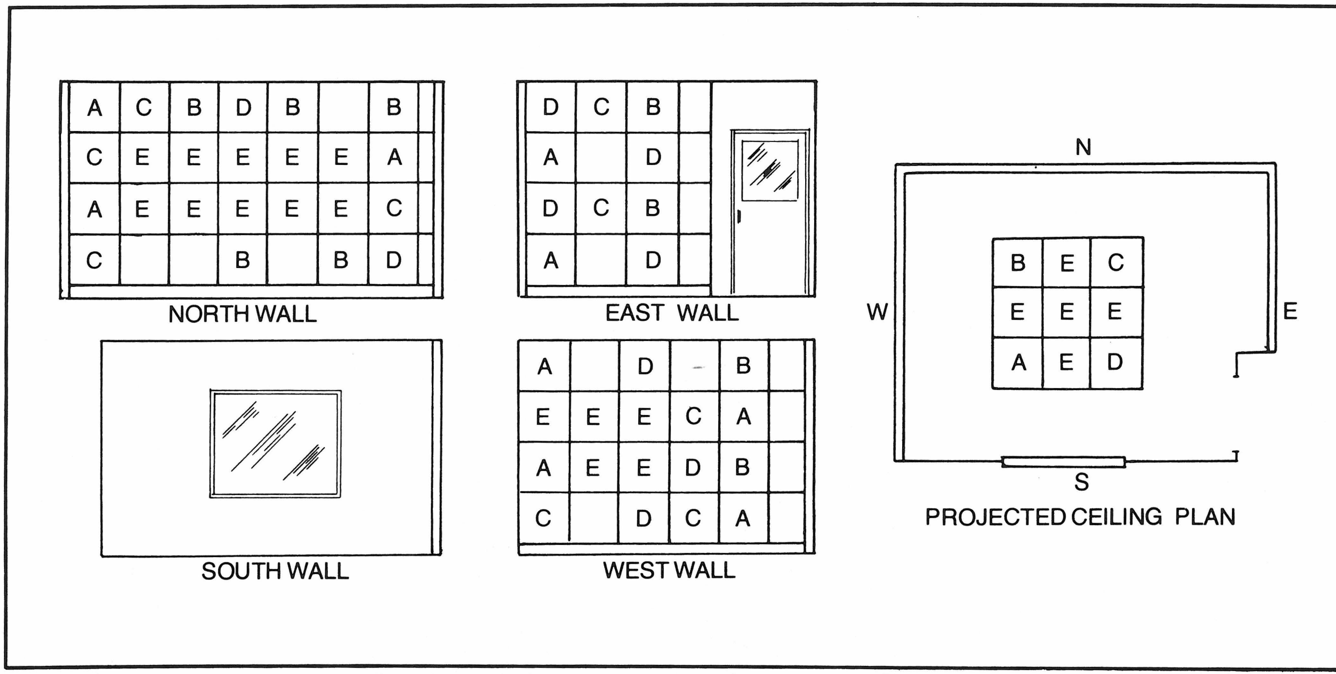


Fig. 11-3. A grid of lightweight 4″×1-¾″ steel channel forms wall modules of 24″×24″ size. The absorption characteristics of each module is determined by the perforation percentage of its cover. The letter designations are keyed to Figs. 11-4 and 11-5. Modules without letter designations may be covered with blank panels or left open.

provides a reasonably flat reverberation time characteristic of about 0.3 second.

DRYWALL ABSORPTION

It is ironic that for the most commonly used material in studio construction (drywall) absorption coefficients are most difficult to come by. Many sources give the absorption coefficients for ½″ gypsum board on 2×4's, 16″ on centers according to the first entry in Table 11-2. The second entry (from APBA 1975 Bulletin) has greatly reduced absorption at 125 Hz, even though the AMA (reference 1a) and AIMA (reference 1b) are predecessor associations to APBA (reference 2). This is one problem, the lack of agreement on coefficients for the identical ½″ gypsum board material.

Another problem is that other thicknesses and multiple thicknesses of drywall are often used and no coefficients are available for these. Double layers of ⅝″ gypsum board are used in this studio. What coefficients should be used? When all walls and ceilings are of gypsum board, the area is great and the difference in absorption, especially at 125 Hz, is a dominant factor. For example, Studio A has a gypsum board area of 626 sq ft. A coefficient of 0.29 gives 181 sabins, a coefficient of 0.10 only 62 sabins. The difference of

Table 11-2. Drywall Sound Absorption.

Material	Frequency-Hz 125	250	500	1k	2k	4k	Ref.
½″ Gypsum board nailed to 2×4s 16″ on centers, painted	0.29	0.10	0.05	0.04	0.07	0.09	1
½″ Gypsum board nailed to 2×4s 16″ on centers, painted	0.10	0.08	0.05	0.03	0.03	0.03	2

Reference 1a Acoustical Materials Association, Bulletin XXVIII, 1968

1b Acoustical and Insulating Materials Association, Bulletin XXXI, 1971-72

1c United States Gypsum, brochure SC-811/USG/74

1d Compendium of Materials for Noise Control, 1978 and 1980

2 Acoustical and Board Products Association, Bulletin, 1975

Table 11-3. Reverberation Calculations for Studio-A.

Material	S Sq. Ft.	125 Hz		250 Hz		500 Hz		1 kHz		2 kHz		4 kHz	
		a	Sa	a	Sa	a	Sa	a	Sa	a	Sa	a	Sa
Carpet	176	0.01	1.8	0.05	8.8	0.10	17.6	0.20	35.2	0.45	79.2	0.65	114.4
Gypsum board	626	0.10	62.6	0.06	37.6	0.05	31.3	0.04	25.0	0.07	43.8	0.09	56.3
(10) Type A Module	40	1.0	40.0	0.68	27.2	0.35	14.0	0.27	10.8	0.20	8.0	0.20	8.0
(10) Type B Module	40	0.62	24.8	1.0	40.0	0.68	27.2	0.35	14.0	0.27	10.8	0.20	8.0
(10) Type C Module	40	0.20	8.0	0.62	24.8	1.0	40.0	0.68	27.2	0.35	14.0	0.27	10.8
(10) Type D Module	40	0.10	4.0	0.20	8.0	0.62	24.8	1.0	40.0	0.68	27.2	0.35	14.0
(20) Type E Module	80	0.84	67.2	1.0	80.0	1.0	80.0	1.0	80.0	1.0	80.0	1.0	80.0
Total absorption, sabins			208.4		226.4		234.9		232.2		263.0		291.5
Reverberation Time, seconds		0.33		0.31		0.30		0.30		0.27		0.24	

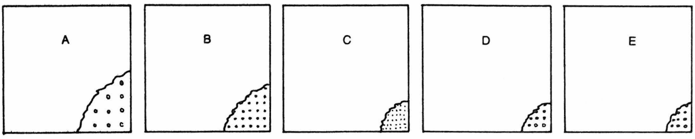
119 sabins is half the total 232 sabins required. It is obvious that precise calculations are in no way possible with such an uncertainty in coefficients.

The absorption of sound by gypsum board is by a flexural, diaphragmatic action, a resonant system. The frequency of resonance (at which peak absorption would be expected) is given by Eq. 9-1. Surface mass of ½″ gypsum board is 2.1 lbs/sq ft and for double ⅝″ gypsum board it is 5.3 lbs/sq ft. From Eq. 9-1 the ½″ gypsum board with 3.75″ airspace resonates at 60.6 Hz and the double ⅝″ gypsum board resonates at 38.1 Hz. With the double ⅝″ material the resonance point is almost 2 octaves below 125 Hz, our first computation frequency. If a peak absorption coefficient such as 0.29 is moved to lower frequencies, the 125 Hz value is correspondingly lowered. All this discussion is to explain the dilemma of every designer of audio spaces and to justify the estimated coefficients used in upcoming calculations.

DISTRIBUTION OF ABSORPTION

From the Sabine equation (Eq. 10-1) the total absorption required to achieve a reverberation time of about 0.3 second may be estimated: (0.049) (1,424)/SA = 0.3 second, the total absorption Sa = 232 sabins. Table 11-3 displays the calculations for Studio A. It is evident that a significant portion of the 232 total required sabins is provided by the carpet and drywall alone. The remainder of absorbing materials to be added will be confined to the 24″ × 24″ modules shown in Fig. 11-3. The five types of absorbers, A, B, C, D, and E, to be fitted into these standard modules are described in Fig. 11-4, four tuned Helmholtz absorbers tuned as shown in Fig. 11-5 and a wideband absorber. With the exception of Type D cover, all types utilize the full 4″ airspace which is filled with glass fiber semi-rigid boards of 3 lb/cu ft density as shown in Fig. 11-4. To utilize the Aslux panel predrilled for 28% perforation, the 1 kHz absorbers require a depth of 2″ instead of 4″. The effective depth can be readily reduced by inserting a panel of ⅛″ hardboard over a styrofoam or other filler under the 2″ glass fiber before the covers are put in place.

The values of reverberation time computed in Table 11-3 are plotted in Fig. 11-6. The slight droop in reverberation time at 2 and 4 kHz is quite acceptable, if it truly exists. Calculations of this type are not precise enough to confirm such details. The uncertainties in absorption coefficients, variabilities in mounting, etc., destroy high precision. The next step is to make measurements, and these



PANEL TYPE	PANEL THICKNESS	HOLE DIA.	HOLE PITCH*	PERFORA-TION %	INSTRUCTIONS
A	1/4"	1/4"	2.71"	0.67	HAND DRILL ASLUX PANEL
B	1/4"	1/4"	1.37"	2.6	HAND DRILL ASLUX PANEL
C	1/4"	1/4"	0.84"	6.9	ASLUX PRE-DRILLED
D	1/4"	11/32"	0.58"	28.	ASLUX PRE-DRILLED-2"DEPTH
E	1/4"	11/32"	0.58"	28.	ASLUX PRE-DRILLED

*Center to center

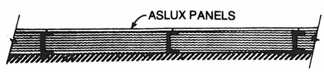


Fig. 11-4. Specifications for the module covers used in Studios A, B, and C. All are tuned Helmholtz resonator absorbers except that module type E is a wide-band absorber. The absorption characteristics of A through D are shown in Fig. 11-5.

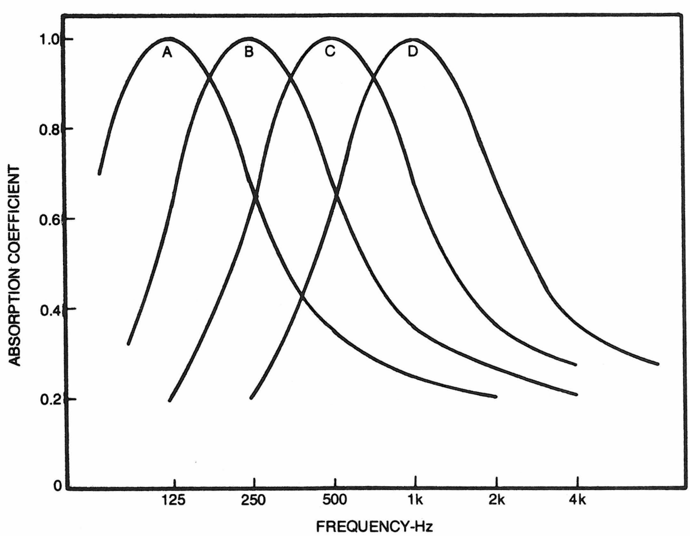


Fig. 11-5. Absorption characteristics of modules A through D described in Fig. 11-4. Type E is a wideband absorber.

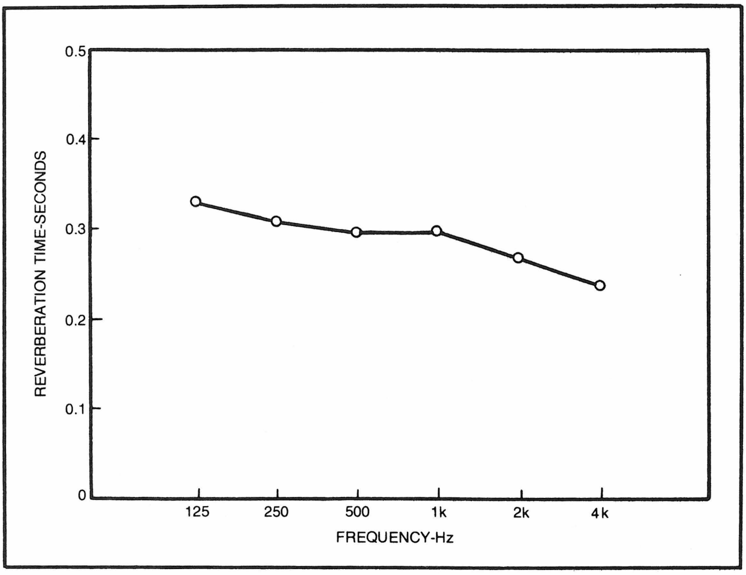


Fig. 11-6. Calculated reverberation time of Pacific Broadcasting Association's Studio A.

should be followed by trimming. The module approach gives very great flexibility in trimming without structural or other renovation.

STUDIO-C TREATMENT

The music studio, Studio C, as shown on the floor plan of Fig. 11-1, has a volume of 5,154 cu ft. Like Studios A and B, it is a room-within-a-room. Section B-B of Fig. 11-1, shown in Fig. 11-7, describes the floating floor and wall construction. The ceiling of double ⅝″ gypsum board is supported resiliently from the building structure as in the small studios.

In establishing the proportions and dimensions of Studio C the distribution of axial frequencies were again the basis of decision. The final dimensional ratios are 1.73 : 1.62 : 1.00 which fall within the Bolt area of Fig. 7-1, but do not coincide with any of the three Sepmeyer ratios of Table 7-2. Table 11-4 gives the axial mode distribution of the room dimensions selected, 22.74 × 19.19 × 11.81 ft. Aside from two near coincidences (1.9 Hz space) the distribution of modal frequencies is satisfactory. For the small Studio A the average separation of axial modal frequencies is 15.51 Hz as shown in Table 11-1. For Studio C, with almost 3½ times the volume, the average separation of modal frequencies (Table 11-4) is down to 10.13 Hz. The lowest axial mode associated with the

longest dimension of the room is 24.8 Hz for Studio C and 37.7 Hz for Studio A. This means that the low frequency response of the larger room is superior to that of the small room. Summary: the larger the room the better the low frequency response, the axial modes are closer together and extend to lower frequencies.

DISTRIBUTION OF MODULES

The reverberation calculations for Studio C are shown in Table 11-5. A goal of 0.6 second reverberation time has been selected. A total of 50 Type B modules (Fig. 11-4) and 25 each of C and D modules are distributed as in Fig. 11-8. Although this calculated reverberation time as plotted in Fig. 11-9 does not conform exactly to the 0.6 second goal, there is no justification in straining further at the calculation level. The imprecise coefficients and assumptions can only give rough guidance. The next step is to measure the reverberation time vs. frequency of the room, after which trimming can be quite precise.

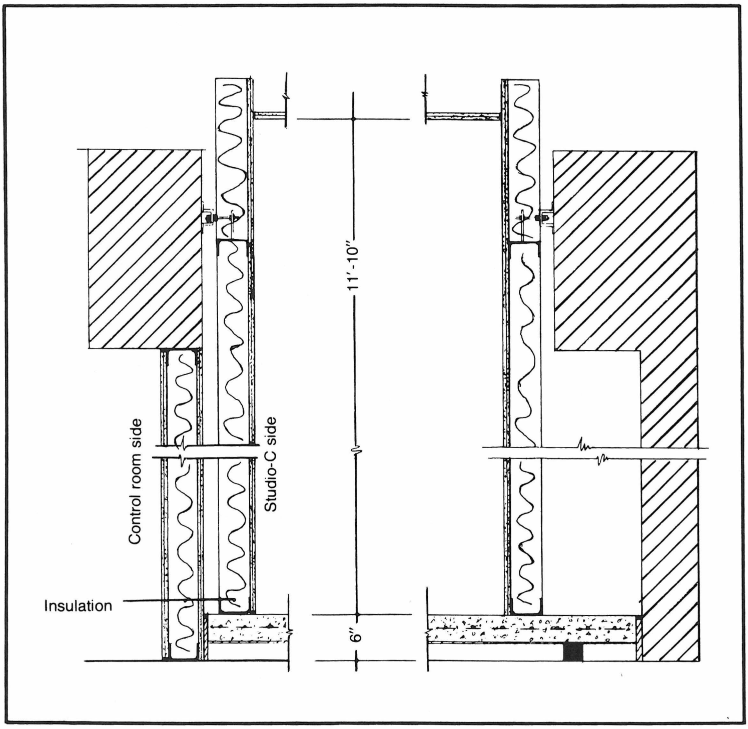


Fig. 11-7. Section B-B of Fig. 11-1 showing wall, ceiling, and floating floor construction of PBA Studio C.

CLOUDS

The ceiling height in Studio C approaches 12 ft. To avoid a cavernous and barnlike impression stepping into this room and to give a more intimate feeling, four "clouds", each 6.5 ft (2 meters) on a side, were specified to be hung from the ceiling as shown in Fig. 11-8. These clouds (Fig. 11-10), made up from lightweight 2″ × 4″ channel welded at the corners, are suspended from the structural rather than the studio ceiling. Because of the small projected area of the cloud frame, vibrations of the building conducted through the wires to the clouds should radiate an insignificantly low sound energy into the room. The channels opening toward the inside of the

Table 11-4. Axial Modes in Studio C.

Room Dimensions: Length . . . . 22.74 ft
Width . . . . . 19.19 ft
Height . . . . 11.81 ft

Axial Mode Resonances				Arranged in Ascending Order	Axial Mode Spacing Hz
	L	W	H		
f_1	24.8	29.4	47.8	24.8	4.6
f_2	49.7	58.9	95.7	29.4	18.4
f_3	74.5	88.3	143.5	47.8	1.9
f_4	99.4	117.8	191.4	49.7	9.2
f_5	124.2	147.2	239.2	58.9	15.6
f_6	149.1	176.7	287.0	74.5	13.8
f_7	173.9	206.1	334.9	88.3	7.4
f_8	198.8	235.5		95.7	3.7
f_9	223.6	265.0		99.4	18.4
f_{10}	248.5	294.4		117.8	6.4
f_{11}	273.3	323.9		124.2	19.3
f_{12}	298.2			143.5	3.7
f_{13}	323.0			147.2	1.9
				149.1	24.8
				173.9	2.8
				176.7	14.7
				191.4	7.4
				198.8	7.3
				206.1	17.5
				223.6	11.9
				235.5	3.7
				239.2	9.3
				248.5	16.5
				265.0	8.3
				273.3	13.7
				287.0	7.4
				294.4	3.8
				298.2	

VOLUME = 5,154 cu ft

MEAN SEPARATION OF AXIAL MODES = 10.13 Hz

STD. Deviation = 6.34 Hz

Table 11-5. Reverberation Calculations for Studio C.

Material	S Sq. Ft.	125 Hz		250 Hz		500 Hz		1 kHz		2 kHz		4 kHz	
		a	Sa	a	Sa	a	Sa	a	Sa	a	Sa	a	Sa
Carpet	436	0.01	4.4	0.05	21.8	0.10	43.6	0.20	87.2	0.45	196.2	0.65	283.4
Gypsum Board	1,426	0.10	142.6	0.06	85.6	0.05	71.3	0.04	57.0	0.07	99.8	0.09	128.3
(50) Type B Module	200	0.62	124.0	1.0	200.0	0.68	136.0	0.35	70.0	0.27	54.0	0.20	40.0
(25) Type C Module	100	0.20	20.0	0.62	62.0	1.0	100.0	0.68	68.0	0.35	35.0	0.27	27.0
(25) Type D Module	100	0.10	10.0	0.20	20.0	0.62	62.0	1.0	100.0	0.68	68.0	0.35	35.0
Total absorption, sabins			301.0		389.4		412.9		382.2		453.0		513.7
Reverberation time, seconds		0.84		0.65		0.61		0.66		0.56		0.49	

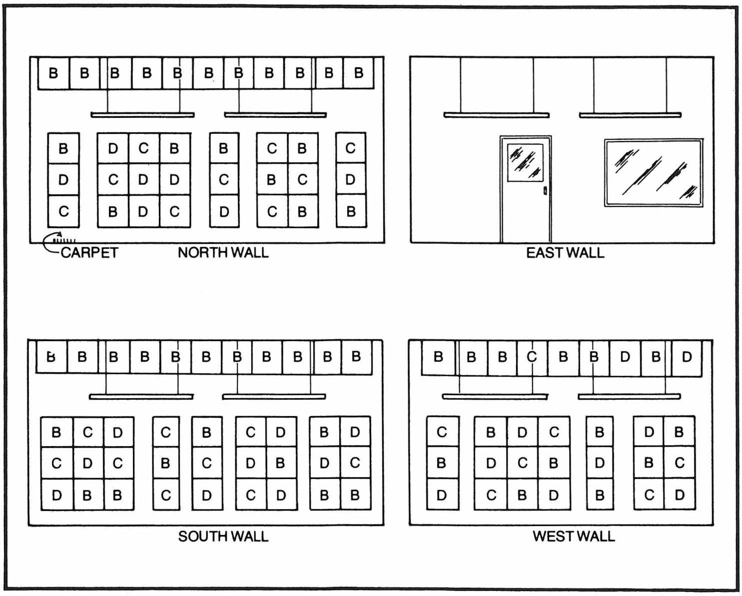


Fig. 11-8. Wall treatment of PBA Studio C for music recording. The letter designations refer to module types of Figs. 11-4 and 11-5.

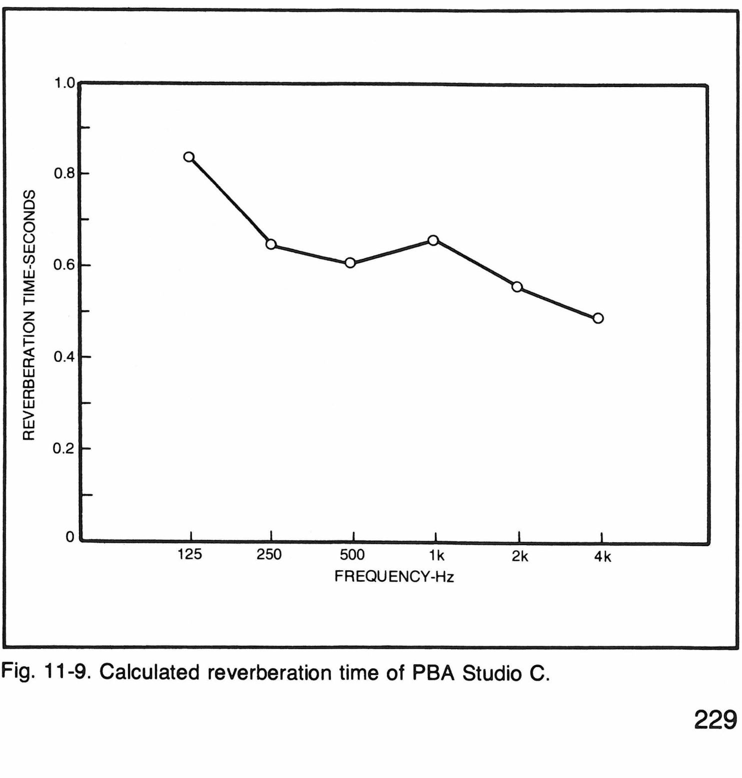


Fig. 11-9. Calculated reverberation time of PBA Studio C.

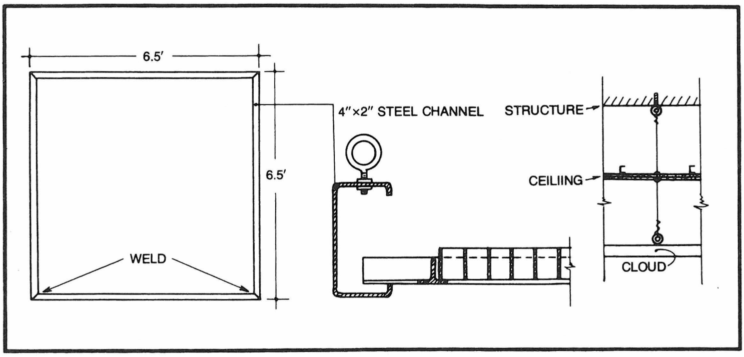


Fig. 11-10. Constructional features of the suspended clouds of Studio C which are designed to provide a virtual visual ceiling at about the 8 ft level for a more intimate effect than that of the actual 12 ft ceiling. Cloud contribution is esthetic, not acoustical.

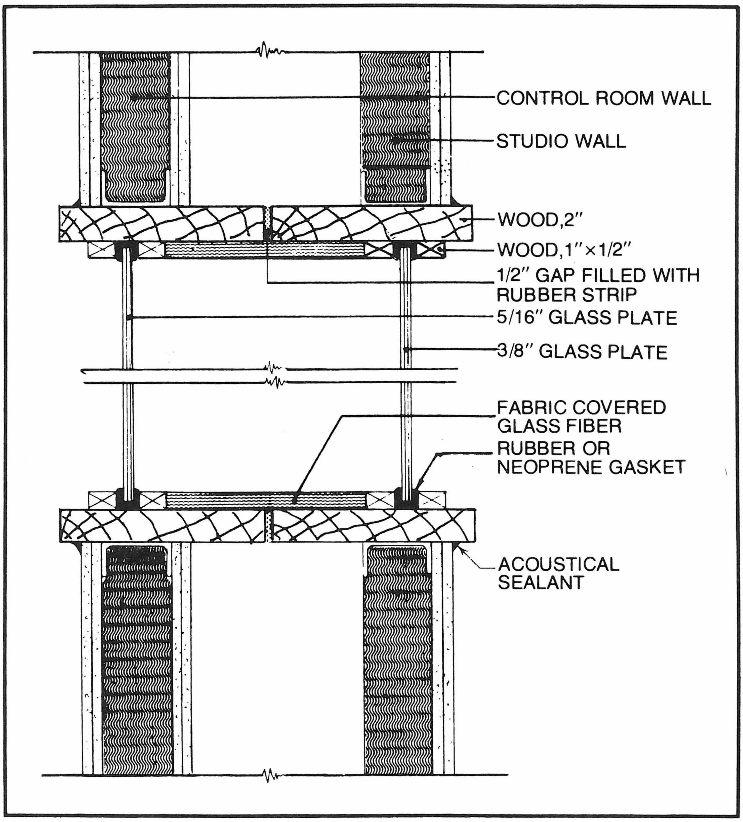


Fig. 11-11. Observation window constructional details to give a window having a transmission loss approaching that of the wall in which it is set.

frame support inverted T-sections which, in turn, support one of the many types of louvered panels which would be wide open acoustically. Illumination fixtures may be accommodated in the clouds. By directing the light downward at the cloud level, glare in the observation window is minimized. If the upper walls and ceilings are painted flat black, upper level service runs are essentially invisible. Such clouds are of esthetic value only and are acoustically neutral.

OBSERVATION WINDOW

A simple, but satisfactory method of constructing the observation window between studio and control room is shown in Fig. 11-11. With a two-leafed wall structure it is important to keep the two faces of the observation window mechanically isolated from each other.

Note added in press—

Actual reverberation measurements made after the above was written and the studios completed yielded the following results:

	Reverberation time, seconds	
	Goal	Actual*
Studio A	0.3	0.271
Studio B	0.3	0.280
Studio C	0.6	0.642

*Average of 7 octave measurements

In general, greater low frequency absorption from drywall and less high frequency absorption from carpet was found than anticipated. With the modular plan it was quite straightforward to adjust the shape of the reverberation time-frequency characteristic of each room to conform to design goals.

12

Acoustical Design of a Control Room

Popular attention is directed primarily to the studio. That is where the name personalities "do their stuff", that is where the action is. The technically oriented person, however, may find greater interest in what transpires on the other side of the glass. The control room is presently the scene of some quite revolutionary changes[1]. New microprocessor-controlled instrumentation has revealed the helpfulness of some sound reflections and the damaging effect of others. Psychoacoustical research is undergirding our understanding of the response of the ear-brain mechanism to sound reflections delayed various amounts. A new day appears to be dawning in control room design, accompanied by wide divergence of opinions and overshadowed by a measure of controversy. Minds are being agitated, traditional approaches are being questioned—in short, it is a very stimulating and healthful atmosphere which is bound to bring solid advance in our approach to control room acoustics.

PURPOSE OF THE CONTROL ROOM

What is the basic purpose of the control room of a recording studio? Needed are a top quality electronic chain, effective electro-acoustic transducers (loudspeakers), and an uncolored acoustical environment. The purpose for these three links in the listening chain is to enable the operator to accurately evaluate sound from the studio uncluttered by control room faults. Surely, any adjustments

the operator makes to correct for control room flaws amount to degradation of the program material.

REVERBERATION TIME IMPORTANCE

There is a fairly widespread appreciation of control room reverberation time. It must be shorter than that of the studio it serves or the operator cannot hear the studio reverberation. The fact that the two rooms, studio and control room, are coupled electro-acoustically affects the reverberation time of the control room as discussed in Chapter 10. Although reverberation time is still one accepted criterion of acoustical excellence and worthy of attention, other factors must also be carefully controlled.

THE LIVE END-DEAD END™ CONTROL ROOM*

A new direction in control room philosophy was introduced by Don Davis of Synergetic Audio Concepts in 1978[2]. In a Syn-Aud-Con seminar time delay spectrometry[3] equipment was being studied and demonstrated. Comb filter distortions resulting from combination of reflections from nearby surfaces and the direct ray were dramatically demonstrated. The audibility of some of these comb filter effects led Davis to ponder the possibility of making the surfaces at the loudspeaker end of the control room absorptive and the other end reflective to get lower level early reflections (which produce audible comb filter effects) and more higher level later reflections (which produce inaudible comb filter effects)[4,5]. One person in that seminar, Chips Davis (not related), went home and applied the idea of an absorptive front end to the control room of his recording studio and the Live End-Dead End™ or LEDE™ control room principle was launched.

THE INITIAL TIME DELAY GAP

As one listens to recorded music there are certain cues the ear-brain system uses to judge the size of the spaces in which the music originates. In a large concert hall the direct sound ray from the musician reaches a listener in the audience and, after a short period of time, dominant reflections from walls, etc. arrive. The gap between the two, known as the "initial time delay" provides the cue by which room size can be judged, as well as other beneficial effects. In a recording studio this initial time delay gap is short because reflections from nearby surfaces are picked up by the microphone

*Live End-Dead End™ and LEDE™ are trademarks of Synergetic Audio Concepts of San Juan Capistrano, CA.

which is also nearby. It is important that the operator in the control room hear the initial time delay gap of the studio in the sound from the monitor loudspeakers. In the traditional control rooms this is not possible because the initial time delay gap of the studio is masked by that of the control room. In addition to the comb filter effects, Don Davis soon realized the salutary effect of the Live End-Dead End™ principle in also making it possible for the operator to hear the initial time delay gap of the studio for the first time[6].

Highly sophisticated microprocessor-based measuring equipment is now available for measuring the pattern of reflected energy at the operator's position which clearly delineates the initial time delay gap. In the Live-End-Dead End™ control room the initial time delay gap is maintained by suppressing early reflections from nearby surfaces and favoring reflection 15 milliseconds or so later from the reflective rear of the room.

THE FORMAL LEDE™ CONTROL ROOM

Don Davis has registered the terms "Live End-Dead End" and "LEDE" to limit their use to rooms which are properly designed to achieve the objectives discussed above and whose characteristics have been completely verified by qualified consultants using time-energy-frequency measuring equipment. Upon such verification, permission to use the Live End-Dead End™ and LEDE™ terms in describing the control room is given with no fee. The intention is to focus attention on the basic technical requirements and to avoid dilution of their importance.

THE INFORMAL CONTROL ROOM

The control room to be described does not meet the formal requirements and is definitely not to be called a Live End-Dead End™ or LEDE™ room. To qualify it as such is impractical from the budget standpoint, foreign location, etc. However, with full acknowledgment of Don Davis' contribution to our understanding of control room theory and practice, information from his published papers has been applied to this control room with the expectation of achieving at least partial benefits in the listening environment.

CONTROL ROOM A

Control Room A serves Studio A as shown in the floor plan of Fig. 11-1. As an economy measure no floating floors or special walls

and ceiling are used in the control room. The wood floor is raised to provide space for the myriad of signal cables and is covered with vinyl or other such wearing surface. The exterior (east) wall is concrete, the others drywall.

The ceiling is the underside of the structural floor above, cut up by numerous beams and a soffit containing HVAC ducts. In other words, no special precautions have been taken for protection against structure-borne noise. There is some comfort in knowing that such noise will not be recorded on the outgoing product. It is hoped that such noise will not seriously interfere with evaluation of sound from the monitor loudspeakers.

ABSORPTIVE FRONT

A sectional view of Control Room A is shown in Fig. 12-1 and the wall elevations in Fig. 12-2. The shaded areas represent absorptive treatment of ceiling, floor, and wall areas except the glass of the observation window. This treatment line extends approximately to the operator's ear position, some 5 ft from the front wall.

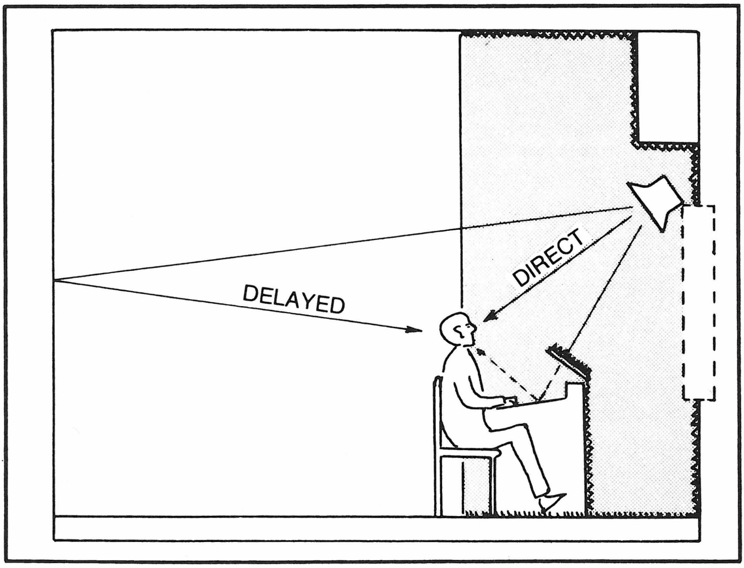


Fig. 12-1. By suppressing short-delay reflections with an absorbent front end, and encouraging relatively long delay reflections from the rear, the initial time delay gap of the control room is made longer than that of the studio, allowing that of the studio to be heard by the control room operator. Other advantages of this distribution of absorbing material include a reduction of audible comb filter colorations.

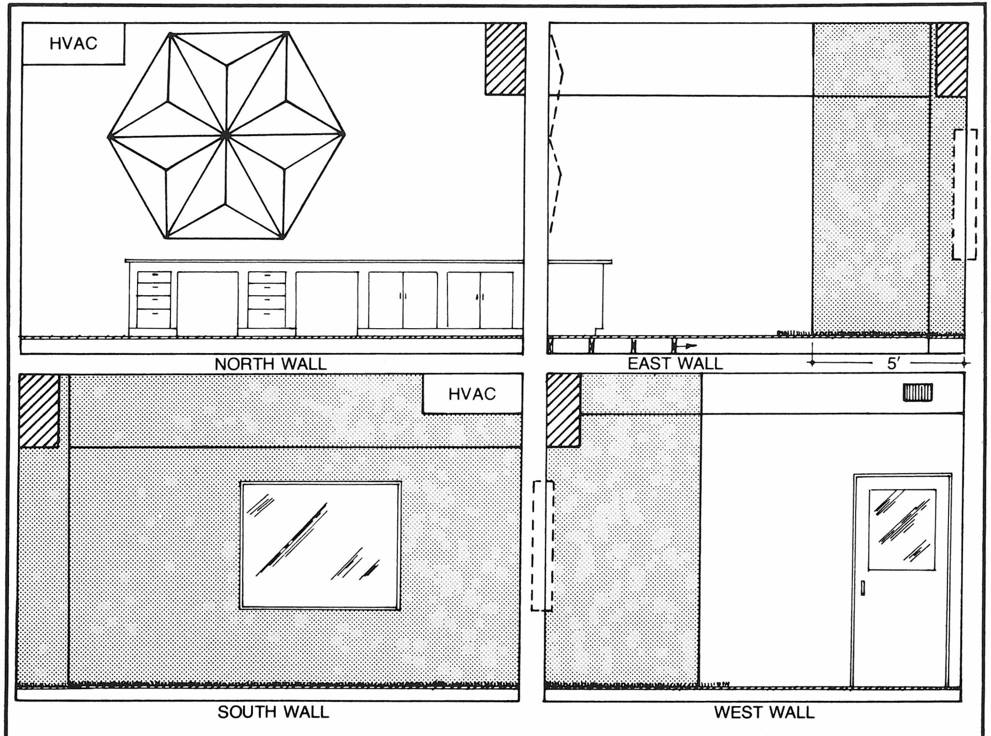


Fig. 12-2. Wall elevations of Control Room A showing distribution of absorbing material and the diffusing element on the north wall.

To keep the operator's chair from rolling on and off the carpet edge, the edge is extended to some practical point. The original intention was to specify 2″ Sonex for wall and ceiling treatment which is easily mounted by cementing. However, fire regulations prohibited it. A locally manufactured product was specified in its place. This is Glasron, a 2″ thick glass fiber board of 4 lb/cu ft density with a glass cloth cover. This board is cemented in place on all the absorptive surfaces but the floor.

REFLECTIVE REAR

The path of sound rays (only geometric frequencies are considered) are idealized in Fig. 12-1. The direct ray travels from loudspeaker to the operator's ears. Rays reflected from the rear wall, traveling greater distances, arrive at the operator's ears some 15 ms or so later than the direct ray, producing an initial time delay gap.

Reflections from the console face are intercepted by a special absorptive shield. Such console reflections, if not reduced, produce short delay comb filter distortions resulting in audible colorations. The longer delay reflections from the rear also produce comb filters, but their null and peak spacings (see Chapter 1) are so close in frequency as to be inaudible.

DIFFUSER

The rear (north) wall of Control Room A provides some diffusion from the work table surfaces. This incidental diffusion is augmented by the geometric diffuser[7] detailed in Fig. 12-3. It appears to be a complicated structure but, in reality, is very simple to construct. Squares of ½″ plywood 2′-8″ on a side are cut diagonally. Three of these triangles (1, 2, 3) are assembled to make one pyramid which is affixed to the wall.

Six such pyramids complete the pyramidal diffusing element. Painted rising sun colors will assure maximum visual impact (if that is what is desired). The acoustical impact is that sound rays falling on this pyramidal element are directed to reflective rear walls, ceiling, and floor and thence to the operator's ears, giving a torrent of reflections of varying delays but all arriving after the initial time delay gap.

REVERBERATION TIME

The wooden floor of Control Room A is made of ¾″ plywood on 2 × 6's. This is a diaphragmatic low frequency absorber as is the

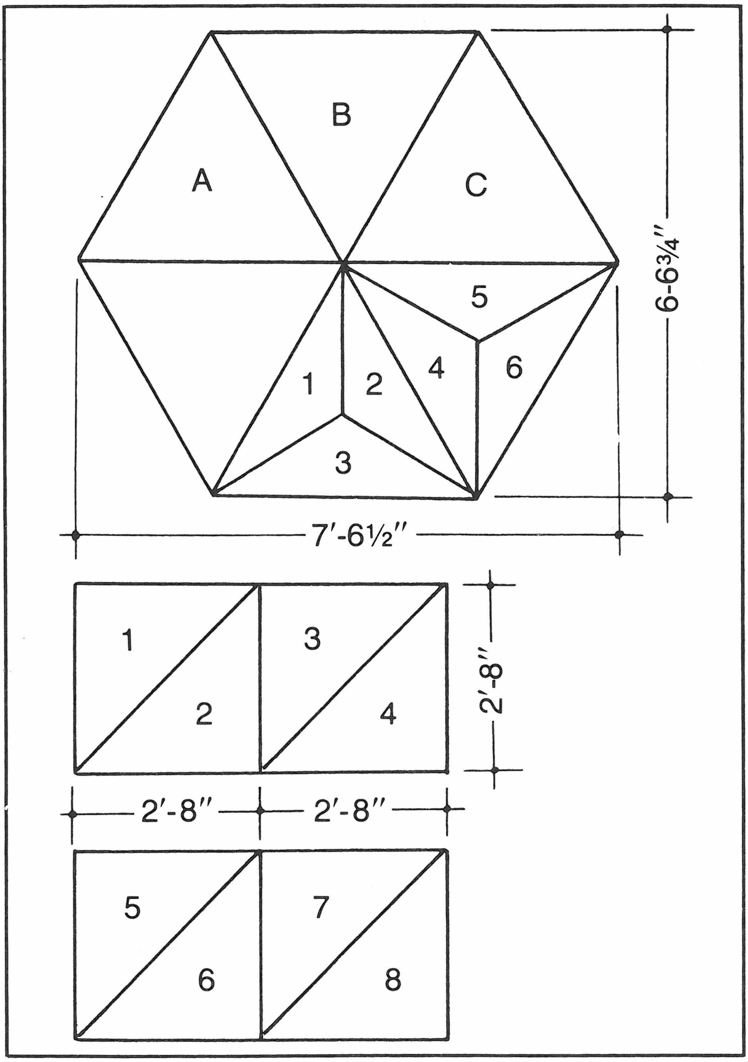


Fig. 12-3. Steps in the construction of the north wall diffusing element. Triangular pieces of ½″ plywood 1, 2, and 3 are mounted to the wall to form a single pyramid. Six such pyramids complete the diffuser.

drywall surface. The carpet and 2″ Glasron of the absorptive front end are excellent absorbers at higher audio frequencies, but deficient at low frequencies. The wooden floor and drywall tend to compensate for the carpet and Glasron absorption giving a reverberation time of about 0.25 second to serve the 0.3 second reverberation time of Studio A.

WHAT IS THE OPERATOR'S IMPRESSION?

A cleaned up sound results from reduction of short delay reflections from the console face and nearby surfaces. The absorptive front end and console face shield have reduced the audible, short delay comb filter colorations.

The time delay between direct sound and sound reflected from the rear of the room allows the operator to hear the initial time delay gap of the studio unmasked by his own.

The Haas effect (Chapter 4) fuses together the stream of delayed reflections from the rear increasing the subjective loudness without perception of the individual reflections as "echoes".

The delayed reflections give the operator the impression of being in a much larger space, a distinct plus.

The above effects combine to provide a clarity of perception of sound unmatched in a traditional control room.

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1. Everest, F. Alton. *The Master Handbook of Acoustics.* Tab Books, Blue Ridge Summit, Pennsylvania, (Especially Chapter 16)
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3. Heyser, R.C. *Acoustical Measurements By Time Delay Spectrometry.* Jour. Audio Engr. Soc., Vol. 15, No. 4 (April 1967), pg. 370.
4. Davis, Don and Chips Davis. *The LEDE™ Concept For The Control Of Acoustic And Psychoacoustic Parameters In Recording Control Rooms.* Jour. Audio Engr. Soc., Vol. 28, No. 9 (September 1980), pp. 585-595.
5. Davis, Don. *Engineering an LEDE™ Control Room For A Broadcasting Facility.* presented at the 67th convention of the Audio Engr. Soc., New York (October-November 1980), Preprint #1688 (I-1).
6. Davis, Don. *The Role of The Initial Time Delay Gap In The Design of Control Rooms For Recording And Reinforcing Systems.* Presented at the 64th Convention of the Audio Engr. Soc., (November 1979), Preprint #1547 (F-3)
7. Downs, J.W. *The Effect of Direct Reflections in Auditoriums.* Sound and Vibration, Vol. 13, No. 6 (June 1979), pp. 18-20.

13

Adjustable Acoustics

If a space is used for only one purpose, it can be treated acoustically with some precision. A multipurpose studio, on the other hand, carries with it compromises, the magnitude of which can be roughly estimated by the difference between the optimum reverberation times for music and speech. However, we must remember that such optimum reverberation times are subjective judgments and cannot be taken as the whole truth.

There is general agreement that reverberation time of small studios should be essentially uniform throughout the audible range but a voice rich in overtones might be better served by a reverberation time curve that droops in the highs. If the reverberation time remains flat above 7 kHz, cornets and trombones tend to sound harsh and the effect can be very distracting to other performers, especially in the smaller studios. A compromise is almost always necessary and experience indicates that compromising in the too-dead direction is preferable.

We tend to think of acoustical materials as something to nail or cement to a surface, but let us look at this more carefully. It is possible to pull a drape having one type of absorption characteristic in front of a surface having quite a different characteristic. Hinged panels, rotating elements, or simple portable elements can alter the absorption and hence the reverberation characteristics of a room. With variable elements it is possible to change a flat reverberation characteristic from one value to another. The actual shape of the

reverberation-frequency graph can be changed with or without a change in average reverberation time.

In any given studio situation, the decision must be made as to how much second-order fussing is justified. Perhaps in most cases a single compromise reverberation characteristic would be considered adequate. However, it is well for us to be aware of the techniques for varying the acoustical quality of an enclosure, many of which are quite straightforward and inexpensive.

DRAPERIES

As radio broadcasting developed in the 1920s, draperies on the wall and carpets on the floor were almost universally used to "deaden" studios. During this time there was remarkable progress in the science of acoustics. It became more and more apparent that the old radio studio treatment was quite unbalanced, absorbing middle- and high-frequency energy but providing little absorption at the lower frequencies. As proprietary acoustical materials became available, hard floors became common and drapes all but disappeared from studio walls.

A decade or two later the acoustical engineers, interested in adjusting the acoustical environment of the studio to the job to be done, turned with renewed interest to draperies. A good example of this early return to draperies was illustrated in the rebuilding of the old Studio 3A of the National Broadcasting Company of New York City in 1946. This studio was redesigned for optimum conditions for making records for home use and transcriptions for broadcast purposes. The acoustical criteria for these two jobs differ largely as to the reverberation-frequency characteristic. By the use of drapes and hinged panels (to be considered later), the reverberation time was made adjustable over more than a two-to-one range. The heavy drapes were lined and interlined and were hung some distance from the wall to make them more absorbent at the lower frequencies. When the drapes were withdrawn, polycylindrical elements having a plaster surface were exposed. (Plywood was in critical supply in 1945-46.)

If due regard is given to the absorption characteristics of draperies, there is no reason other than cost why they should not be used. The effect of the fullness of the drape must be considered (Fig. 9-5). The acoustical effect of an adjustable element using drapes can thus be varied from that of the drape itself when closed (Fig. 13-1) to that of the material behind when the drapes are withdrawn into the slot provided. The wall treatment behind the

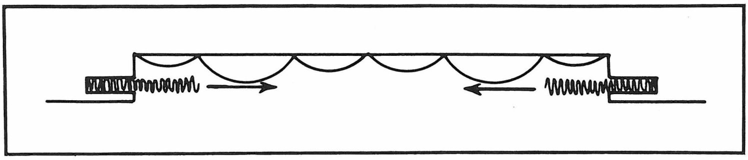

Fig. 13-1. The reverberation of a room may be varied by pulling absorptive drapes in front of reflective areas.

drape could be anything from hard plaster for minimum sound absorption to resonant structures having maximum absorption in the low-frequency region, more or less complementing the effect of the drape itself. Acoustically, there would be little point to retracting a drape to reveal material having similar acoustical properties.

PORTABLE PANELS

Portable absorbent panels offer a certain amount of flexibility in adjusting listening room or studio acoustics. The simplicity of such an arrangement is illustrated in Figs. 13-2(A) and (B). In this example a perforated hardboard facing, a mineral fiber layer, and an air cavity constitute a low frequency resonator. Hanging such units on the wall adds low-frequency absorption, acoustically removes some of the highly reflective wall surface, and contributes somewhat to sound diffusion by its shape. There is some compromising of the effectiveness of the panels as low-frequency resonators in that the units hang loosely from the mounting strip. "Leakage" coupling between the cavity and the room would tend to destroy the resonant effect. Panels may be removed to obtain a "live" effect for instrumental music recording, for example, or introduced for voice recording.

Free standing "acoustical flats" are useful studio accessories. A typical flat consists of a frame of 1 × 4 lumber with plywood back filled with a low density (e.g., 3 lb per cu ft) glass fiber board faced with a fabric such as muslin or glass fiber cloth to protect the soft surface. Arranging a few such flats strategically can give a certain amount of local control of acoustics.

ROTATING ELEMENTS

Rotating elements of the type shown in Fig. 13-3 have been used in radio station KSL in Salt Lake City, Utah. In this particular configuration the flat side is relatively absorbent and the cylindrical diffusing element is relatively reflective. A disadvantage of this system is the cost of the space lost which is required for rotation.

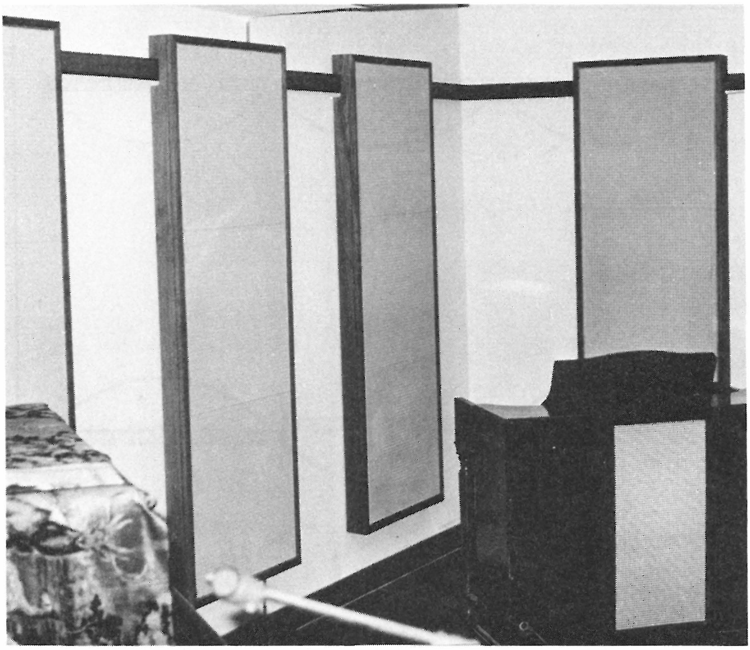

Fig. 13-2 (A). The simplest and cheapest way to adjust the reverberation characteristics of a room is to use removable panels. These photographs were taken in the studios of the Far East Broadcasting Company, Hong Kong.

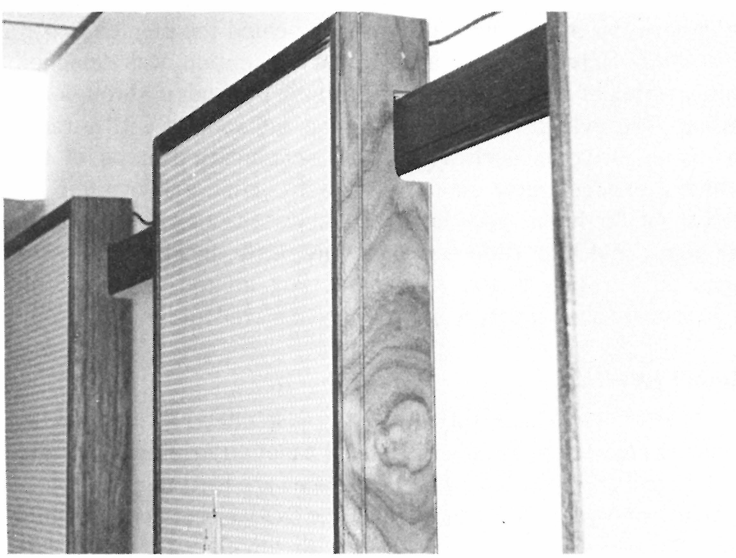

Fig. 13-2 (B). Close-up of hanging detail.

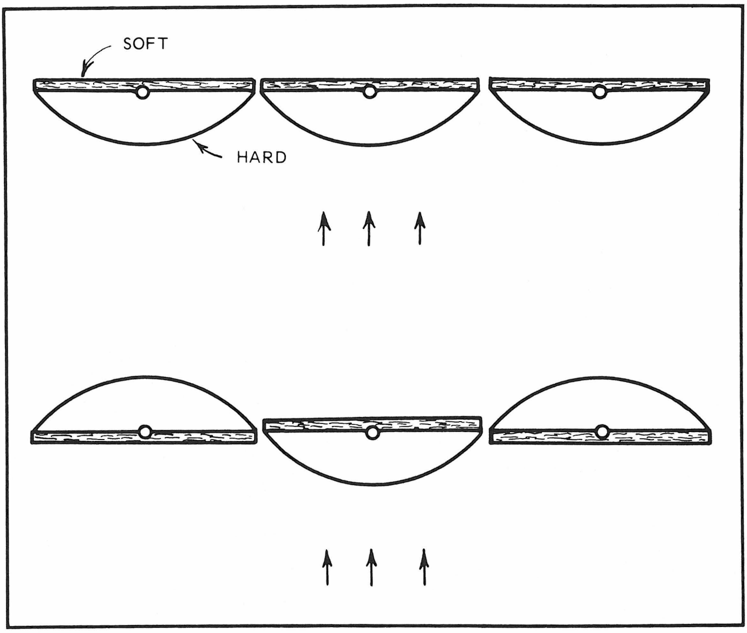

Fig. 13-3. Rotating elements can vary the reverberation characteristics of a room. They have the disadvantage of requiring considerable space to accommodate the rotating elements.

The edges of the rotating element should fit tightly to avoid coupling between the studio and the space behind the elements.

At the University of Washington a music room was designed with a series of rotating cylinders partially protruding through the ceiling. The cylinder shafts are ganged and rotated with a rack-and-pinion drive in such a way that sectionalized areas of the cylinder exposed give moderate low-frequency absorption increasing in the highs, good low-frequency absorption decreasing in the highs, and high reflection absorbing little energy in lows or highs. Such arrangements, while interesting, are too expensive and mechanically complex to be seriously considered for most studios.

HINGED PANELS

One of the least expensive and most effective methods of adjusting studio acoustics is the hinged panel arrangement of Figs. 13-4(A) and (B). When closed, all surfaces are hard (plaster, plasterboard, or plywood). When opened, the exposed surfaces are soft. The soft surfaces may be covered with low-density glass fiber blanket 2 to 4 inches thick. This blanket could be covered with cloth

for the sake of appearance. Spacing the glass fiber from the wall would improve absorption at low frequencies.

VARIABLE RESONANT DEVICES

Resonant structures for use as sound absorbing elements have been used extensively in the Danish Broadcasting House in Copenhagen. One studio used for light music and choirs employs pneumatically operated hinged perforated panels as shown in Fig. 13-5. The effect is basically to shift the resonant peak of absorption as shown in Fig. 13-5(B). The approximate dimensions applicable in Fig. 13-5(A) are: width of panel 2 ft, thickness ⅜ in., holes ⅜ in. diameter spaced 1-⅜ inch on centers. A most important element of the absorber is a porous cloth having the proper flow resistance covering either the inside or outside surface of the perforated panel.

When the panel is in the open position the mass of the air in the holes and the "springiness" (compliance) of the air in the cavity behind act as a resonant system. The cloth offers a resistance to the vibrating air molecules, thereby absorbing energy. When the panel is closed the cavity virtually disappears and the resonant peak is shifted from about 300 to about 1700 Hz (Fig. 13-5(B)). In the open condition the absorption for frequencies higher than the peak remains remarkably constant out to 5000 Hz.

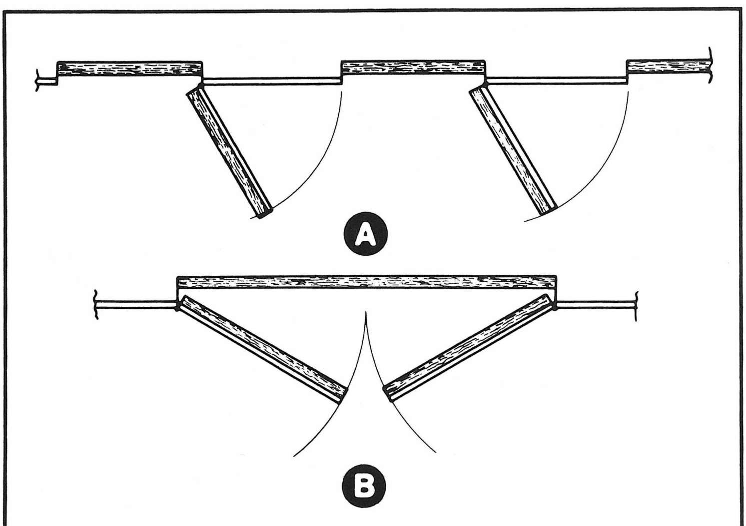

Fig. 13-4. An inexpensive and effective method of incorporating variability in room acoustics is through the use of hinged panels, hard on one side and absorbent on the other.

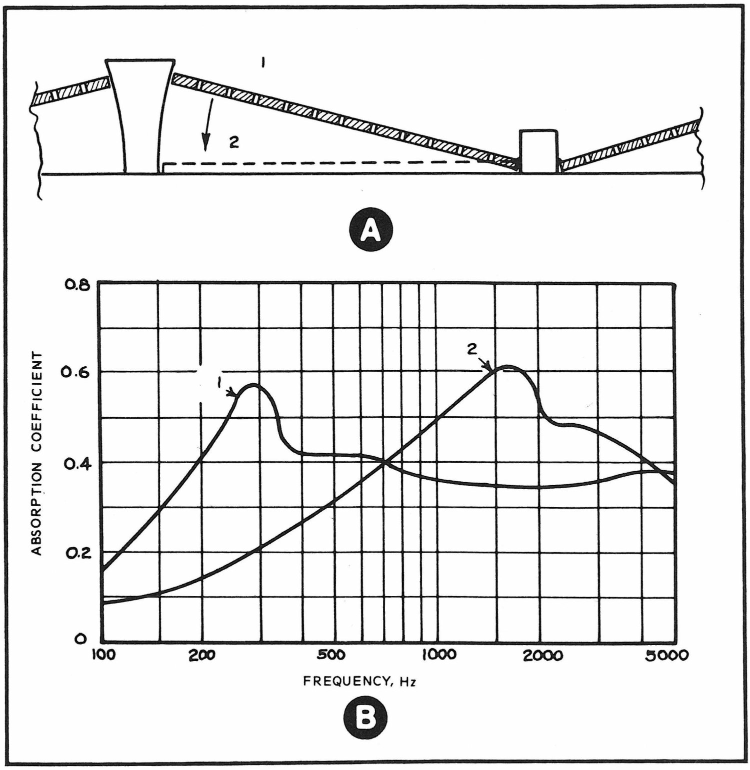

Fig. 13-5. (A) Pneumatically operated hinged, perforated panels used to vary reverberation in the Danish Broadcasting House in Copenhagen. An important element not shown is a porous cloth of the proper flow resistance covering one side of the perforated panel. (B) Changes in absorption realized by shifting the panel of (A) from one extreme to the other.

LOUVERED PANELS

While performing the household chore of washing the louvered windows, the author was struck with the possibility of using louvered panels in front of soft absorbing material for varying the reverberation characteristics of a studio. And then it was discovered that the Danes used this "Venetian Blind" method a quarter of a century ago. The louvered panels of an entire section can be rotated by the action of a single lever in the frames commonly available for home construction, Fig. 13-6(A). Behind the louvers is a low-density glass fiber board or batt. The width of the panels determines whether they form a series of slits, Fig. 13-6(B), or seal tightly together, Fig. 13-6(C). In fact, opening the louvers of Fig. 13-6(C) slightly would approach the slit arrangement of Fig. 13-6(B) acous-

tically, but it might be mechanically difficult to arrange for a precise slit width.

The louvered panel arrangement is basically very flexible. Assuming that the louvers are open, the glass fiber can be of varying thickness and density and fastened directly to the wall or spaced out different amounts. The louvered panels can be of hard material (glass, hardboard) or of softer material such as wood and they can be solid, perforated, or arranged for slit-resonator operation. In other words, almost any absorption-frequency characteristic we have seen in the graphs of earlier chapters can be matched with the louvered structure with the added feature of adjustability.

THE SNOW ADJUSTABLE ELEMENT

Although snow is a phenomenally good sound absorber, it is neither readily available in many parts of the world nor well adapted to listening room or studio applications. We have in mind here a studio design by the late William B. Snow for the sound mixing-looping stage at Columbia Pictures Corporation studios in Hollywood[1]. Sound mixing requires good listening conditions; looping requires variable voice recording conditions to simulate the many acoustic situations of motion picture scenes portrayed. In short, reverberation time had to be adjustable over about a two-to-one range for the 80,000cu. ft.stage.

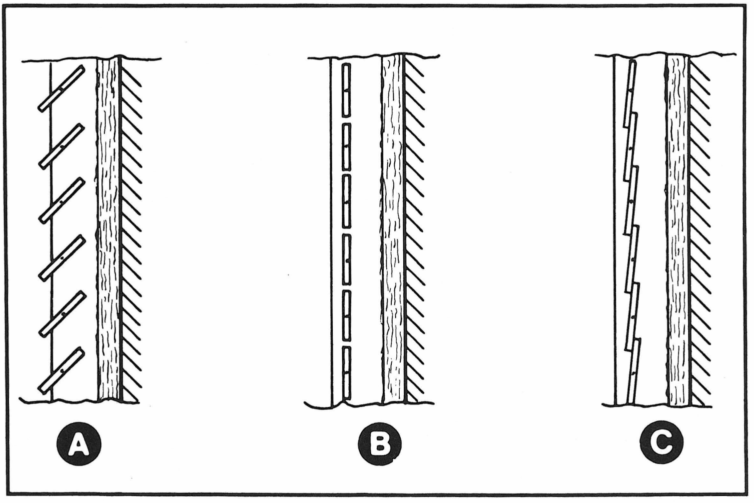

Fig. 13-6. Louvered panels may be opened to reveal absorbent material within, or closed to present a reflective surface. Short louvers can change from a slat resonator (closed) to reveal absorbent material within (open).

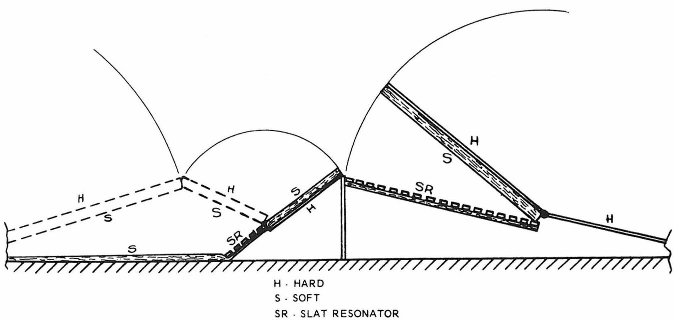

Fig. 13-7. Variable acoustical elements in the mixing-looping stage at Columbia Pictures Corporation, Hollywood. Reflective areas are presented when closed, absorbent areas and slat resonators are presented when the doors are opened. (After Snow[1])

Both side walls of the stage were almost covered with the variable arrangement of Fig. 13-7 which is a cross section of a typical element extending from floor to ceiling with all panels hinged on vertical axes. The upper and lower hinged panels of 12 ft length are hard on one side (2 layers of ⅜ in. plaster board) and soft on the other (4 in. fiberglass). When open they present their soft sides and reveal slit resonators (1 × 3 slats spaced ⅜ in. to ¾ in. with mineral fiber board behind) which utilize the space behind the canted panels. In some areas glass fiber was fastened directly to the wall. Diffusion is less of a problem when only highly absorbent surfaces are exposed but when the hard surfaces are exposed, the hinged panels meet, forming good geometric diffusing surfaces.

The Snow design illustrates the extreme flexibility offered the engineer in combining many types of absorbers in an effective yet inexpensive overall arrangement and there is no reason why the hi-fi enthusiast couldn't use them just as effectively with a bit of care.

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1. Snow, William B. *Recent Applications of Acoustical Engineering Principles in Studios and Review Rooms*. Jour. Soc. Motion Picture and Television Engr., Vol. 70 (January 1961) pp. 33-38.

14

Tuning the Listening Room

A good ear is indispensible to the process of adjusting high fidelity equipment to give optimum results in a specific room. In fact, without good ears, fine tuning a high fidelity installation is the ultimate exercise in futility. Let us assume a good pair of ears in the following discussion. By "good" ears is meant a hearing ability that is reasonably normal. The only way the reader can be assured of possessing normal hearing acuity is to submit to a test procedure administered by a licensed audiologist. From this audiogram comes (1) assurance that hearing is normal within limits, (2) revelation that certain deficiencies exist which can be taken into consideration in equipment and room adjustments, and/or (3) a deeper appreciation of and respect for this fantastic ability which builds a new attitude toward excessive sound levels.

THE SOUND LEVEL METER

In order to tune a listening room it is necessary to have some sort of measuring instrument. In this case it is a sound level meter. Fortunately, there are available sound level meters adequate for our purposes at reasonable prices.[5] These will be described later. First, an introduction to the instrument itself.

A sound level meter is really a very simple device. It consists of a microphone to convert sound levels to voltage levels, an audio amplifier to amplify the voltage levels, frequency weighting networks, a calibrated attenuator to extend the range, and an indicating

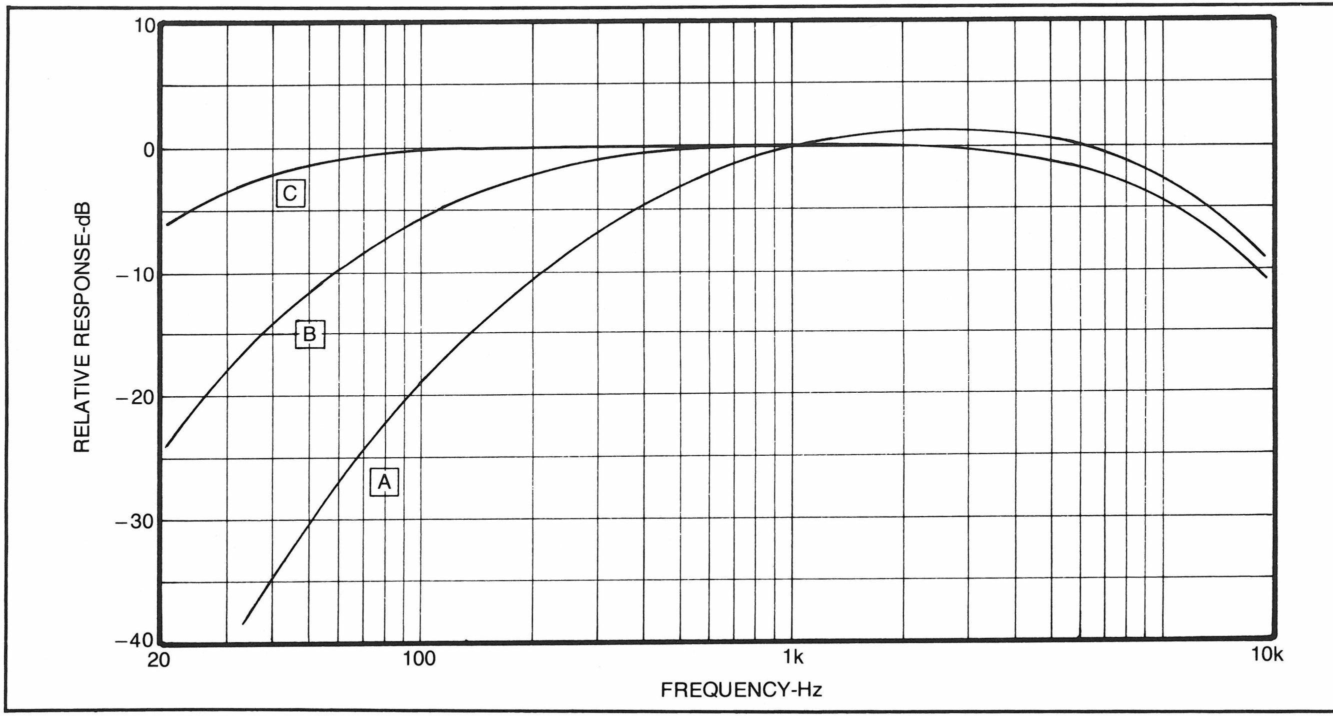

Fig. 14-1. A, B, and C weighting response characteristics for sound level meters. (ANSI S1 .4-1971.)

Table 14-1. Sound Level Meter Random-Incidence Relative Response Level as a Function of Frequency for Various Weightings*

Frequency Hz	A Weighting Relative Response dB	B Weighting Relative Response dB	C Weighting Relative Response dB
10	−70.4	−38.2	−14.3
12.5	−63.4	−33.2	−11.2
16	−56.7	−28.5	−8.5
20	−50.5	−24.2	−6.2
25	−44.7	−20.4	−4.4
31.5	−39.4	−17.1	−3.0
40	−34.6	−14.2	−2.0
50	−30.2	−11.6	−1.3
63	−26.2	−9.3	−0.8
80	−22.5	−7.4	−0.5
100	−19.1	−5.6	−0.3
125	−16.1	−4.2	−0.2
160	−13.4	−3.0	−0.1
200	−10.9	−2.0	0
250	−8.6	−1.3	0
315	−6.6	−0.8	0
400	−4.8	−0.5	0
500	−3.2	−0.3	0
630	−1.9	−0.1	0
800	−0.8	0	0
1000	0	0	0
1250	+0.6	0	0
1600	+1.0	0	−0.1
2000	+1.2	−0.1	−0.2
2500	+1.3	−0.2	−0.3
3150	+1.2	−0.4	−0.5
4000	+1.0	−0.7	−0.8
5000	+0.5	−1.2	−1.3
6300	−0.1	−1.9	−2.0
8000	−1.1	−2.9	−3.0
10 000	−2.5	−4.3	−4.4
12 500	−4.3	−6.1	−6.2
16 000	−6.6	−8.4	−8.5
20 000	−9.3	−11.1	−11.2

*ANSI S1.4-1971′.

meter. Another helpful adjunct, not always present, is an output connector to accommodate additional measuring equipment. Obviously, the quality of the instrument depends on the quality of the integral parts, sturdiness of construction, and stability of calibration.

Microphones used for high quality sound recording are generally inadequate for sound level meter use. Stability, uniformity of response, and omnidirectional characteristics are important, the

latter requiring a microphone of small size. The amplifier is typically stabilized by heavy inverse feedback.

WEIGHTING NETWORKS

Weighting networks are incorporated in an attempt to make sound level meter readings meaningful with respect to human perception of loudness. The three most commonly used weighting networks are A, B, and C as shown in graphical form in Fig. 14-1, with a flat or linear scale added in some instruments. In Table 14-1 the response of the A, B, and C weighting networks are shown in tabular form. The A weighting network is the approximate inverse of the 40 phon equal loudness contour of Fig. 4-4.

In like manner the B and C weighting networks are the approximate inverse of the 70 and 100 phon contours. It follows that A weighting is intended for use in measuring low sound levels, B weighting for intermediate sound levels, and C weighting for loud sounds.

SOUND LEVEL METER TYPES

Sound level meters are used for a wide range of measurements, some requiring high precision, others lower precision. The four standard types are:

Type 1 - Precision
Type 2 - General Purpose
Type 3 - Survey
Type S - Special Purpose

The tighter the tolerances, the more expensive the instrument. An example of the range of tolerances for Types 1, 2, and 3 is given in Table 14-2 for the C weighting condition.

TYPICAL SOUND LEVEL METERS

A few of the many less expensive sound level meters are described in Table 14-3. For measurements in the home listening room to be described in this chapter, sound level meters of Type 2 or 3 are usually quite adequate as well as Type S, especially if a flat response is available. Several typical sound level meters of the low and medium price ranges are shown in Fig. 14-2.

THE LISTENING CHAIN

The elements of all high fidelity listening systems are shown in

Fig. 14-3. First is the source of the signals to which we listen. This can be a phonograph pickup for sounds recorded in disc form. It can be a cassette or reel-to-reel magnetic reproducer. It could even be a live speech or musical source except that this would merge into a sound reinforcement setup which is outside the scope of this book. Next comes the equipment such as preamplifiers, power amplifiers, and the electroacoustical transducer, the loudspeaker. If there is signal processing equipment such as equalizers, limiters, compressors, or reverberators, it belongs in this category.

The third element in Fig. 14-3 is room acoustics which couples the output of the loudspeakers to the human hearing system. This is

Table 14-2. Total Tolerance Limits for Sound Level Meters*.

FREQ.-Hz	Type 1	Type 2	Type 3
10	±2.5		
12.5	±2		
16	±2		
20	±2	+3.0, −∞	+4,0 −∞
25	±1.5	±2.0,−2.5	+3.0,−4.5
31.5	±1.5	+1.5,−2.0	+2.5,−3.0
40	±1	+1.0,−1.5	+2.0,−2.5
50	±1	±1.0	±2.0
63	±1	±1.0	±2.0
80	±1	±1.0	±2.0
100	±1	±1.0	±2.0
125	±1	±1.0	±2.0
160	±1	±1.0	±2.0
200	±1	±1.0	±2.0
250	±1	±1.0	±2.0
315	±1	±1.0	±2.0
400	±1	±1.0	±2.0
500	±1	±1.0	±2.0
630	±1	±1.0	±2.0
800	±1	±1.0	±2.5
1000	±1	±1.5	±2.5
1250	±1	±1.5	±2.5
1600	±1	±2.0	±3.0
2000	±1	±2.5	±3.5
2500	±1	+3.5,−3.0	±4.0
3150	±1	+4.5,−3.5	±4.5
4000	±1	+5.0,−4.0	±5.0
5000	+1.5,−2	+5.5,−4.5	±6.0
6300	+1.5,−2	+6.0,−5.0	±7.0
8000	+1.5,−3	+6.0,−6.0	±7.0
10 000	+2, −4	+6.0,−∞	+7.0,−∞
12 500	+3, −6		
16 000	+3, −∞		
20 000	+3, −∞		

*ANSI S1.4-1971[1]

Table 14-3. Specifications for Typical Sound Level Meters Suitable for Acoustical Response Measurements of Listening Rooms.

Manufacturer Model	Type	Weighting Network & Response	Sound Level Range (re 20 Pa)	Microphone	Features
Scott[6] Model 450B	Type 3	A,B,C fast/slow	30-140 dB	Rochelle salt crystal, diaphragm	External calibration External input External output
Scott[6] Model 452	Type 2	A,B,C fast/slow	35-140 dB	Miniature Ceramic	Acoustical calibration External input External output Mic. extension cable available
Quest[7] Model 215	Type 2	A,B,C Linear fast/slow	30-140 dB	PZT ceramic	Octave filters available Acoustical calib. avail, Mic. extension cable available
Realistic (Radio Shack)	Type S	A,C fast/slow	60-126 dB		
Hall Engineering[4] Model SLM-201	Type S	A, Flat fast/slow	60-126 dB	On 9″ probe	Individually calibrated for both free and semi-free fields.
Pulsar[8] Model 40	Type 2	A,B,C fast/slow	40-135 dB	1″ condenser	Logarithmic amplifier 35 dB range on linear scale, output jack, Mic. extension cable available

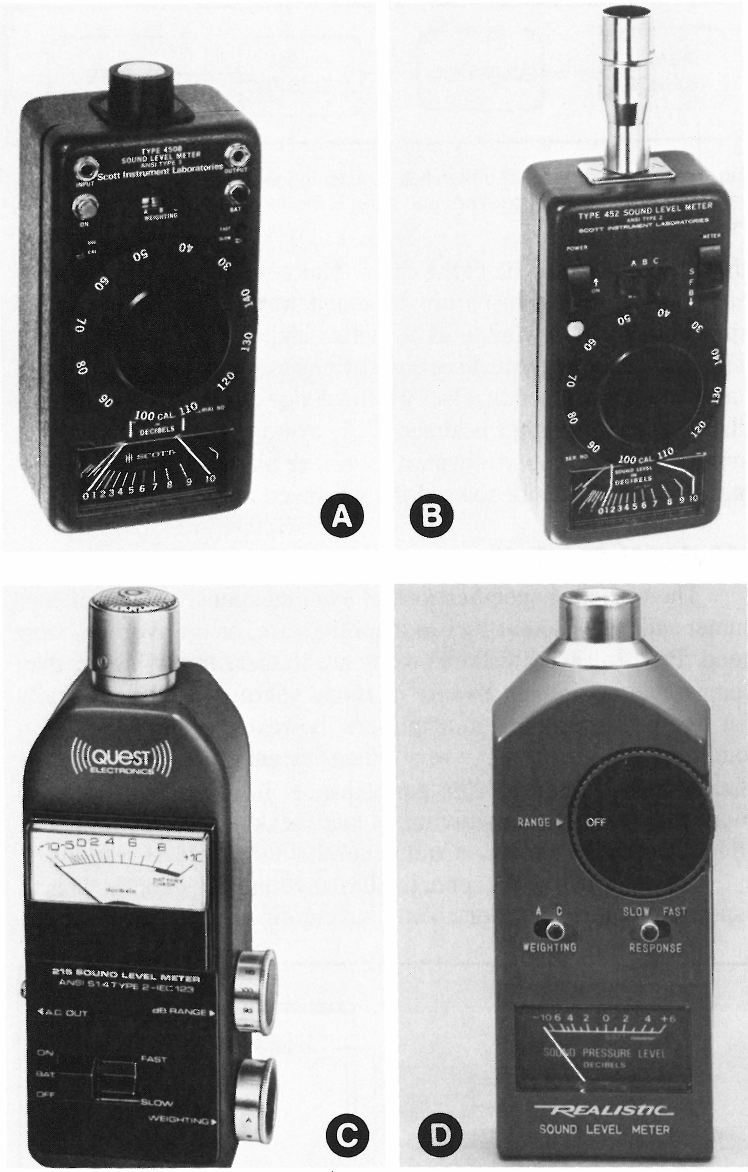

Fig. 14-2. Typical sound level meters suitable for acoustical response measurements in listening rooms (refer to Table 14-3): (A) Scott Model 450-B, a medium priced Type 3 instrument, (B) Scott Model 452, a more accurate Type 2 instrument somewhat more expensive than the 450-B, (C) Quest Model 215, a Type 2 instrument of upper medium price range, and (D) The Realistic sound level meter, the least expensive, available at Radio Shack stores. Useful in many tasks in which accuracy and stability are secondary.

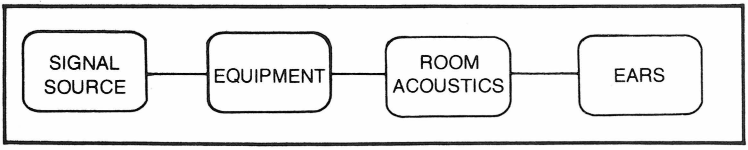

Fig. 14-3. The elements of a high fidelity listening system. The room acoustical link, interfaced by loudspeakers and ears, is the least known and most widely neglected.

the most neglected link of the chain. This neglect may be due to the ephemeral, intangible nature of sound waves, and/or the widespread lack of knowledge of acoustics and acoustical techniques. The intangibility of sound seems destined to stay with us as a part of nature, but it is hoped that lack of knowledge can be at least partially dispelled. This chapter is aimed at the average home hi-fi listening system while the next chapter, covering more serious measurements, is aimed more toward the advanced audio rooms.

ACOUSTICAL RESPONSE

The technical specifications of even inexpensive lines of consumer audio equipment such as amplifiers are, on the average, very good. Performance standards today are immeasurably higher than comparable equipment twenty or thirty years ago. The generally flat frequency response in amplifiers, however, is not equalled in loudspeakers, although the very expensive units are flatter than the cheaper ones. Loudspeaker performance is inseparably tied to room acoustics. The positioning of loudspeakers in a room greatly affects their bass response and general smoothness of response.

The response of the room itself is anything but smooth. This is especially true of smaller rooms having dimensions of the same

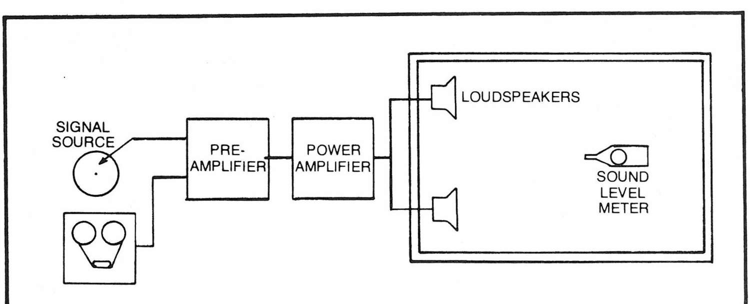

Fig. 14-4. Diagram for measuring the acoustical response of a high fidelity listening system. The sound level meter is placed at the position normally occupied by the listener's ears.

Fig. 14-5. A typical living room listening setup in which the acoustical response is being evaluated. (Brüel & Kjaer Instruments, Inc.)

order of magnitude as the sound being considered. The combination of axial, tangential, and oblique modes determines the room response at a given position and this response is very much a function of frequency and position in the room.

A measurement of the "acoustical response" of a high fidelity listening room gives an overall response embracing phonograph pickup, amplifiers, loudspeakers, and room acoustics. A convenient method of obtaining the acoustical response of a room is diagrammed in Fig. 14-4. Test signals from a special disc recording or other source are played into the system. These signals, reproduced as sounds in the room under test, are picked up by a sound level meter whose microphone is located at the listener's position as shown in Fig. 14-5. This test does not include the listener's hearing mechanism, but it does supply information on the sound field falling on the ears, the rest being left to the aforementioned audiogram.

THE B&K PINK NOISE TEST RECORD

The test record QR 2011 offered by Brüel & Kjaer of Naerum, Denmark[2] is designed for testing and adjustment of hi-fi equipment in the place where it is to be used, whether in the home or dealer's demonstration room. The test should be first conducted with all bass, midrange, or treble settings in either their neutral or usual settings as a point of departure. The volume control, however, must be a bit higher than normal as the recording level is 10 dB or so lower than normal.

The contents of this record are given in detail in Table 14-4. Bands 1 and 2 of Side A of this record are a test of the acoustical response of the left channel of the entire system, including room acoustics. Bands 3 and 4 are the same for the right channel. Side B, bands 1 and 2 repeat side A bands 1 and 2 except both loudspeakers are energized. While the signals on side A of the record and bands 1 and 2 of side B are geared primarily to manual taking of data, side B band 3 has only 5 second signals for each band for use with a graphic level recorder. Side B band 4 is broadband pink noise designed to check the polarity of loudspeaker wiring. The signal during the first 30 seconds is correctly polarized which is followed by 30 seconds of signal polarized in the opposite way. If the system under test is connected properly, the loudspeaker diaphragms move in and out in unison and the first 30 second noise is heard between the loudspeakers. If they are incorrectly connected the noise seems to come from all sides. Reversing the leads to one of the loudspeakers

Table 14-4. Contents of Brüel & Kjaer Pink Noise Test Record QR 2011.

	Band	Duration	Channel	Signal Type	Purpose of Test
SIDE A	1	60 sec	Left	⅓ octave band of pink noise centered on 1 kHz	Calibration
	2	500 sec	Left	⅓ octave bands of pink noise 20 Hz-20 kHz (voice identification of center frequency)	Measure Left Channel
	3	60 sec	Right	⅓ octave band of pink noise centered on 1 kHz	Calibration
	4	500 sec	Right	⅓ octave bands of pink noise 20 Hz-20 kHz (voice identification of center frequency)	Measure Right Channel
SIDE B	1	60 sec	Left + Right	⅓ octave band of pink noise centered on 1 kHz.	Calibration
	2	500 sec	Left+ Right		Synchronization For Level
	3	150 sec	Left+ Right		Synchronization For Level Recorder
	4	60 sec	Left+ Right	Broadband pink noise 20Hz-20 kHz	Polarity check
	5	40 sec	Left+ Right	Noise 20 Hz-1 kHz Noise 1 kHz-4 kHz Noise 4 kHz-20 kHz	Bass polarity Midrange polarity Treble polarity
	6	85 sec	Left+ Right	Sine sweep 20 Hz-20 kHz	Check for rattles
	7	240 sec	Left+ Right	Constant broadband pink noise 20 Hz-20 kHz	Subjective listening room response

will correct the problem if one is revealed by this test. Side B band 5 does the same with filtered noise to energize the bass, midrange, and treble loudspeaker components one at a time to reveal possible connection errors within the loudspeaker itself. Side B band 6 is a simple sine sweep from 20 Hz to 1 kHz to check for rattles and other mechanically vibrating components of equipment, room treatment, or furnishings which might generate spurious sounds when energized. Side B band 7 is simply a steady broadband pink noise. This signal plus a practical ear can go far in detecting irregularities of response.

USING THE QR 2011 TEST RECORD

With the sound level meter handheld at the listener position (see Fig. 14-5), and with all controls at their normal or neutral settings for the initial test, we are ready to begin. It is helpful if a second person plots the sound level meter readings as the operator calls them off for each ⅓ octave band of pink noise. Before the pickup arm is lowered to the record, however, it is necessary to determine if corrections must be applied to the sound level meter dB readings called out. If corrections must be applied, the assistant should only record the readings called, applying the corrections after the run is completed.

The QR 2011 test record signals are designed to be read with the sound level meter in the "linear" or "flat" mode. If the sound level meter at hand has no linear or flat setting, corrections must be applied. If the C weighting is used, the corrections to be applied are indicated at the top of the B&K QP 2011 graph paper supplied with the disc. These are modest corrections, the extreme being 7 dB at 20 Hz and 12 dB at 20 kHz and a stretch from 80 Hz to 4 kHz with no correction at all. These corrections supplied by B&K are within 1 dB of the later adopted ANSI standards shown for C weighting in Table 14-1.

If A or B weighting is to be used, much larger corrections are necessary which are listed in Table 14-1. For example, using A weighting a 50.5 dB correction is required at 20 Hz which tells us that the 20 Hz reading would undoubtedly be deep in noise. In fact, using A weighting would probably mean that few readings below 100 Hz would be usable. To be usable, the reading from the test record signal should be at least 10 dB above the ambient noise level for that band. We conclude that the drastic reduction of low frequencies in the A weighting renders this setting undesirable for this type of testing and should be used only in emergencies and with full understanding of the compromise involved.

Using B weighting involves lower corrections than A weighting, but still higher than desirable. With B weighting the correction at 20 Hz is 24 dB and is 11 dB at 20 kHz. Budget sound level meters may come with only A and C weighting, but it would indeed be unusual to find one with only B weighting. For this reason it will not be considered further for this type of service.

TEST RECORD RESULTS

A typical acoustical response for the "as found" condition in a 1,200 cu ft listening room with JBL 4301 loudspeaker is shown in

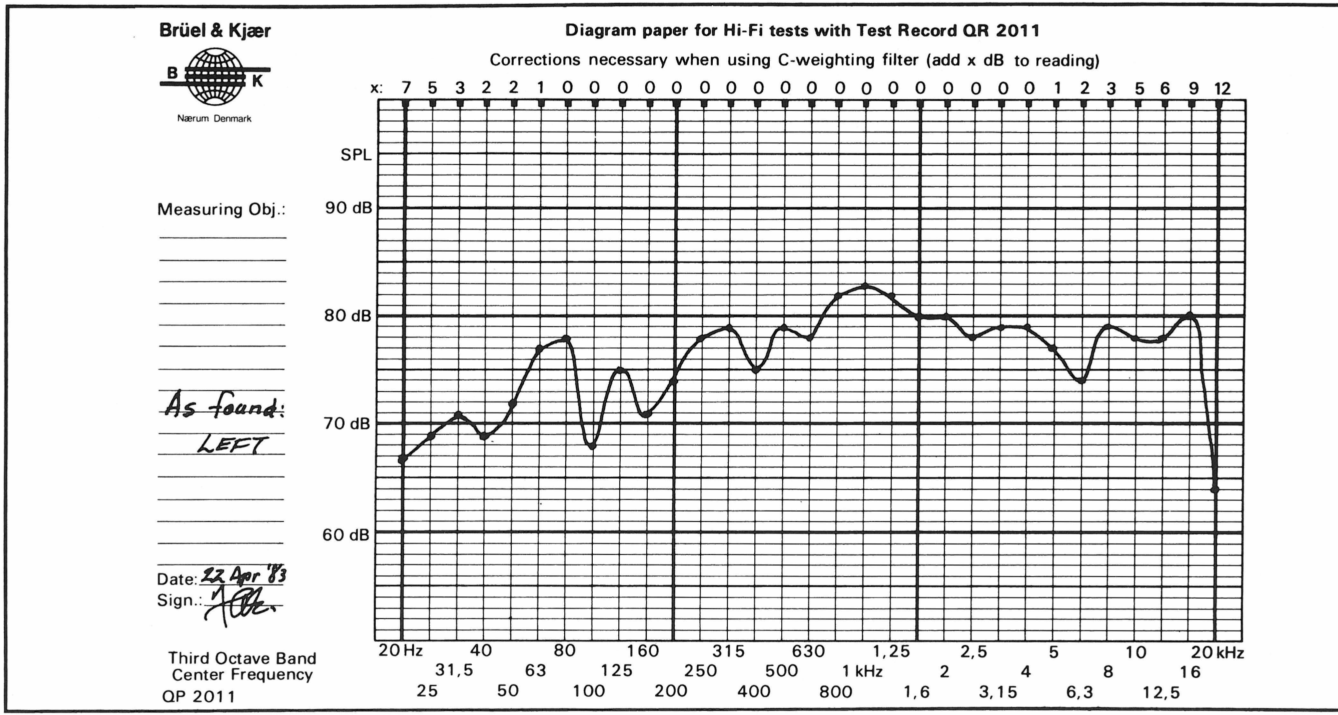

Fig. 14-6. Acoustical response of a high fidelity system in "as found" conditions of loudspeaker placement, equipment settings, etc. Details of this response can be used as the basis of introducing equalization corrections.

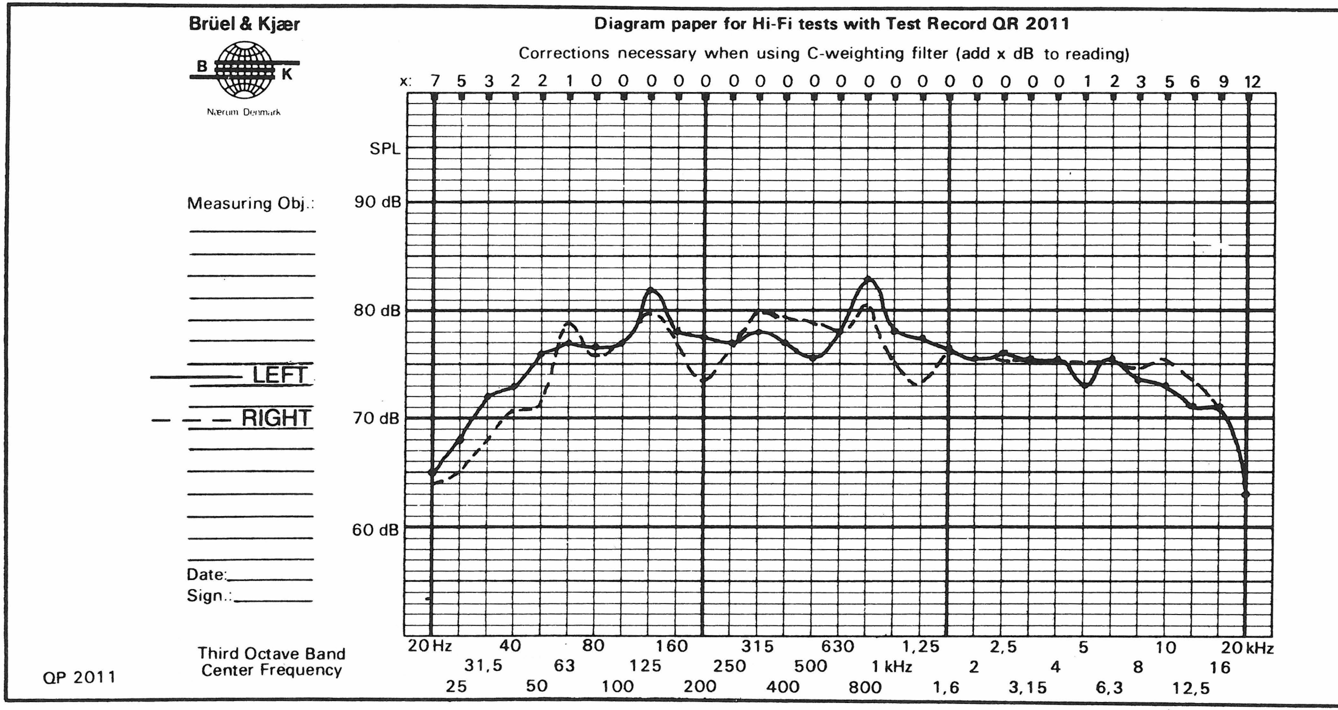

Fig. 14-7. Acoustical response measurements in a 1,700 cu ft control room of a small recording studio following room renovation. These will be the basis of equalization trimming.

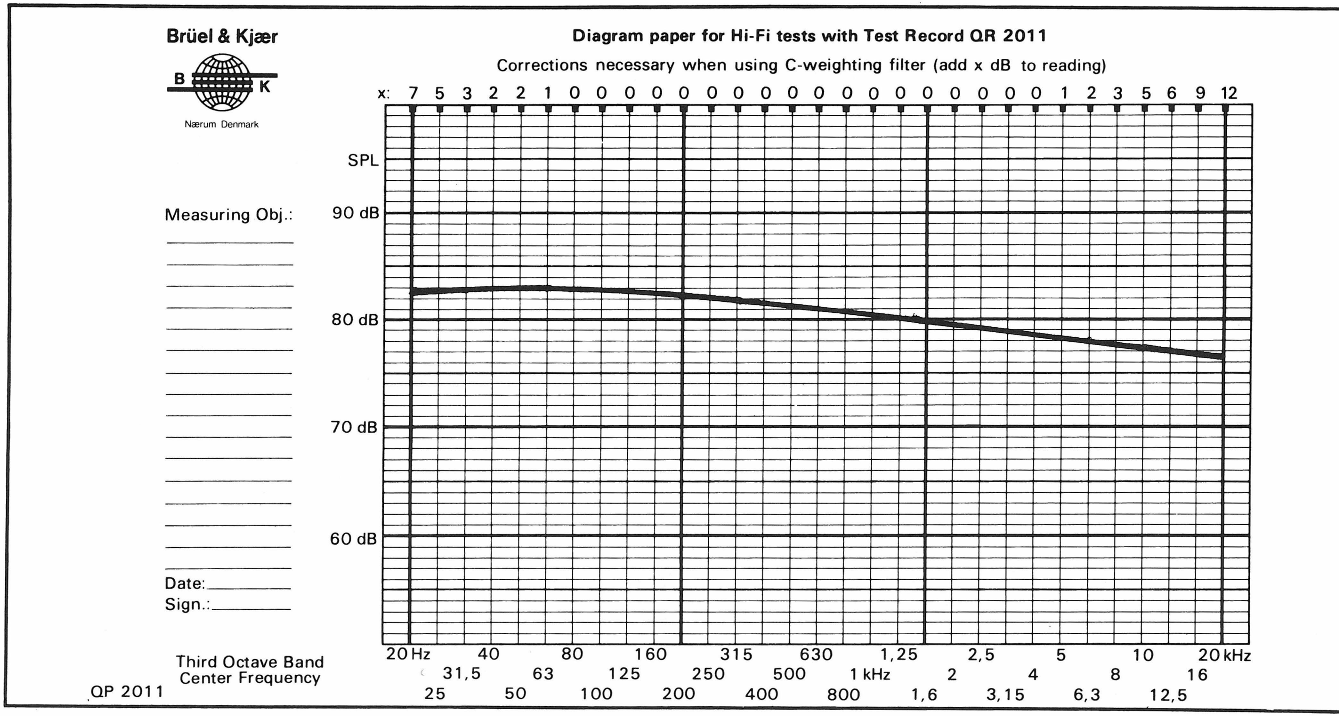

Fig. 14-8. An overall system response recommended by Brüel & Kjaer that is suitable for most recordings of music. For listening to recordings made under far-field conditions a flat response is best. Most modern recordings are made, however, under near-field conditions.

Fig. 14-6. This response curve, measured at the operator's ear position, provides the basis for improvement in loudspeaker mounting and equalization. It was made utilizing the ⅓ octave pink noise bands on side A, band 2 of the QR 2011 test record. The test includes the response of the entire electronic chain, the power amplifier, the loudspeaker, and the acoustics of the room at operator position.

Figure 14-7 shows the acoustical response of left and right monitoring loudspeakers in the 1,700 cu ft control room of a small recording studio. These acoustical response measurements formed the basis of equalization trimming following significant structural changes in the control room.

The instruction booklet accompanying the B&K QR 2011 test record states that the response of Fig. 14-8 is optimum for most music records. For the reproduction of music recorded under far-field conditions, a linear (flat) response is best, but most modern records are made under near field conditions. The response of Fig. 14-8 is a reasonable compromise for average concert halls and average recordings.

FURTHER USES OF THE TEST RECORD

The resources of test recording QR 2011 are far from being fully exploited by using only side A bands 2 and 4. Side B presents signals suitable for level recording tests, polarity checks, and checking for rattles of loose things in the room. The 4 minutes of broadband pink noise on side B band 7 is extremely useful for rapid and convenient subjective judgments. Major flaws can be readily detected by use of this band. All tonal ranges in broadband pink noise are equally powerful. Peaks or dips in the low frequency range, the midband, or high frequency ranges can be detected by ear with familiarization with this signal.

LOUDSPEAKERS VS. ROOMS

The B&K application note[2] gives an excellent example of the measuring procedure just reviewed. Møller measured loudspeakers from five different manufacturers in three different rooms. The conclusion is that the room effect can easily overshadow differences in loudspeakers. It follows that the wise purchaser will select loudspeakers on the basis of which ones sound best in his or her own room, not the dealer's demonstration room.

CBS TEST RECORDS

Another pink noise test record is the STR-140 produced by

Table 14-5. Contents of CBS STR-140 Pink Noise Acoustical Test Record.

SIDE A	
Band 1	Left sweep, 30-15,000 Hz, −14 dB*
Band 2	Right sweep, 30-15,000 Hz, −14 dB*
Band 3	Lateral sweep, 30-15,000 Hz, −14 dB**
Band 4	Left channel spots, 30-14,000 Hz, −14, dB*
Band 5	Left channel reference tone, 1,000 Hz*
Band 6	Right channel reference tone, 1,000 Hz*
Band 7	Lateral reference tone, 1,000 Hz*
SIDE B	
Band 1	Lateral spots, 30-14,000 Hz, −14 dB**
Band 2	Right spots, 30-14,000 Hz, −14 dB*
Band 3	Wideband noise, left channel, −10 dB*
Band 4	Wideband noise, right channel, −10 dB*
Band 5	Wideband noise, left and right channels, polarized properly, −10 dB**
Band 6	Wideband noise, left and right channels, randomly polarized, −10 dB**

* Referred to 3.54 cm/sec rms at 1 kHz
** Referred to 5 cm/sec rms at 1 kHz

CBS Laboratories and available through Columbia Special Products[3]. Its application is quite similar to the B&K record. The contents of this record listed in Table 14-5 are self explanatory. Other records offered by CBS are STR-100 (Stereophonic Frequency Test Record), STR-112 (Square Wave, Tracking, and Intermodulation Test Record), STR-120 (Wide Range Pickup Response Test Record), STR-130 (RIAA Frequency Response Test Record), STR-151 (Broadcast Test Record), STR-170 (318 Microsecond Frequency Response Test Record), STR-101 (Seven Steps To Better Listening), and SQT-1100 (Quadraphonic Test Record). Some bands on these test records are designed to be used with supplemental equipment such as a graphic level recorder.

PHASING VS. POLARITY

If both left and right loudspeaker diaphragms move in and out in perfect unison, it often is said that they are "in phase" or that "the phasing is correct". If one moves outward on a given signal as the other moves inward on the same signal they are said to be "out of phase" and reversing the leads of one of the loudspeakers will correct the situation. Application of the word "phase" in this context, while widespread, is confusing when the larger picture is considered. It is desirable to recognize that phase is frequency dependent, polarity is not. In the loudspeaker case, there is no

frequency dependence if switching a pair of leads corrects the situation, hence polarity is the proper term.

THE HALL SYSTEM FOR ACOUSTICAL TESTING

Hall Engineering[4] offers a relatively inexpensive signal generator which is admirably suited to listening room acoustical response measurements. This is a random noise source which can be used in place of test records, providing a far superior degree of flexibility, precision, and detail. The ATG-301 test generator puts out wideband white noise, wideband pink noise, finite bandwidth pink noise with variable frequency and adjustable bandwidth. Fractional bandwidth pink noise is available in 1/1, 1/2, 1/3, 1/5, 1/10, or 1/20 octave in the range 20 Hz to 20 kHz.

Spectrum analyzers (real-time analyzers or RTAs) come in either of two basic forms. The fractional octave units, commonly ⅓ octave, are quite expensive. The 1/1 octave units are of lower cost and, obviously, much lower in resolution. If the equalizer to be used in smoothing system response is only one octave, surely a one octave real-time analyzer is all that can be justified in determining response. On the other hand, if a ⅓ octave equalizer is to be used, the finer resolution of the ⅓ octave real-time analyzer makes more sense. The Hall ATG-301 noise generator has resolution capabilities far greater than the ⅓ octave analyzers. Its 1/20 octave resolution should be sufficient even to match parametric equalizers and other narrower band equalizers which may be available in the future.

The Hall ATG-301 signal generator, because of its flexibility of adjustment in frequency and bandwidth, is actually a manually operated spectrum analyzer when used with a normal sound level meter. The manual operation is a concession to price and is slow in operation but, at the same time, carries with it advantages the more sophisticated real-time analyzer does not have. It can be set to the exact center frequencies and bandwidths of the equalizer to be used. It offers the freedom of sweeping through the spectrum to locate specific peaks and dips of room response or to explore the frequency region around loudspeaker crossover points.

Any sound level meter can be used with the Hall ATG-301, especially the one Hall offers as a companion instrument, the Model SLM-201. This sound level meter has flat response and A weighting. Its measurement range is 60 dB to 126 dB sound pressure level. The microphone is mounted at the end of a 9-inch probe tube to reduce case diffraction effects.

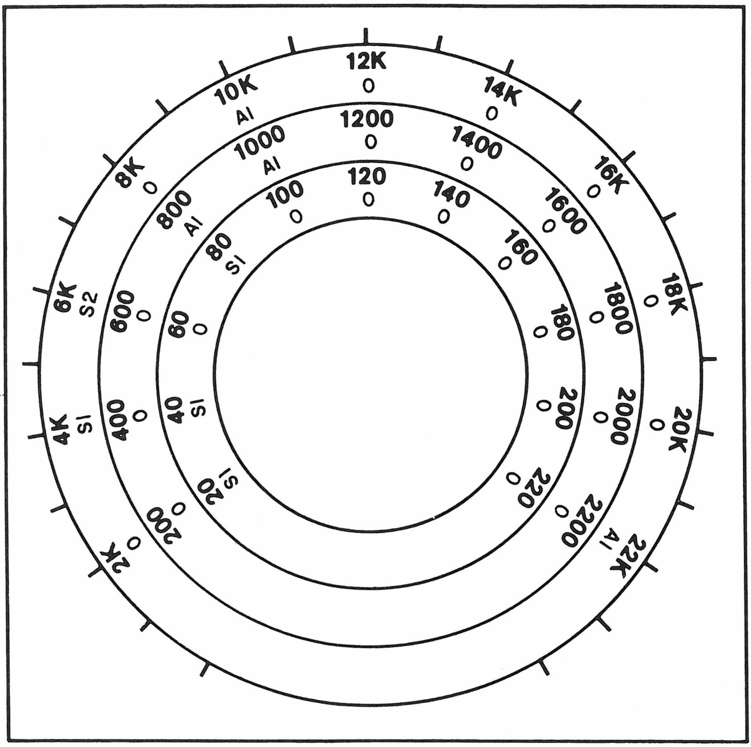

Fig. 14-9. Dial markings on the Hall Engineering ATG-301 noise generator. The small letters under the frequency markings are the calibration corrections (S = subtract, A = add). This generator offers a choice of wideband white noise, wideband pink noise, finite bandwidth pink noise adjustable from 1 octave to 1/20 octave with a frequency range 20 Hz to 20 kHz.

Hall Engineering will supply a calibrated dial for the ATG-301 generator such as that shown in Fig. 14-9 which includes the sound level meter correction to be applied at each frequency. The frequency markings are large black figures, the correction factors are smaller and red in color and are placed immediately below the corresponding frequency marking. The sound level meter calibrations are those fitting the type of sound field in which the instrument is to be used. In the average listening room certainly free field conditions would not prevail. Hall Engineering defines a "normal" field in such locations as a semi-free sound field with 35% diffuse sound on the assumption that between 20% and 50% diffuse sound fields are encountered in most audio testing situations. Accuracy of plus or minus 0.5 dB can be obtained by correcting for the actual percentage of diffuse and free field sound at the location of the sound

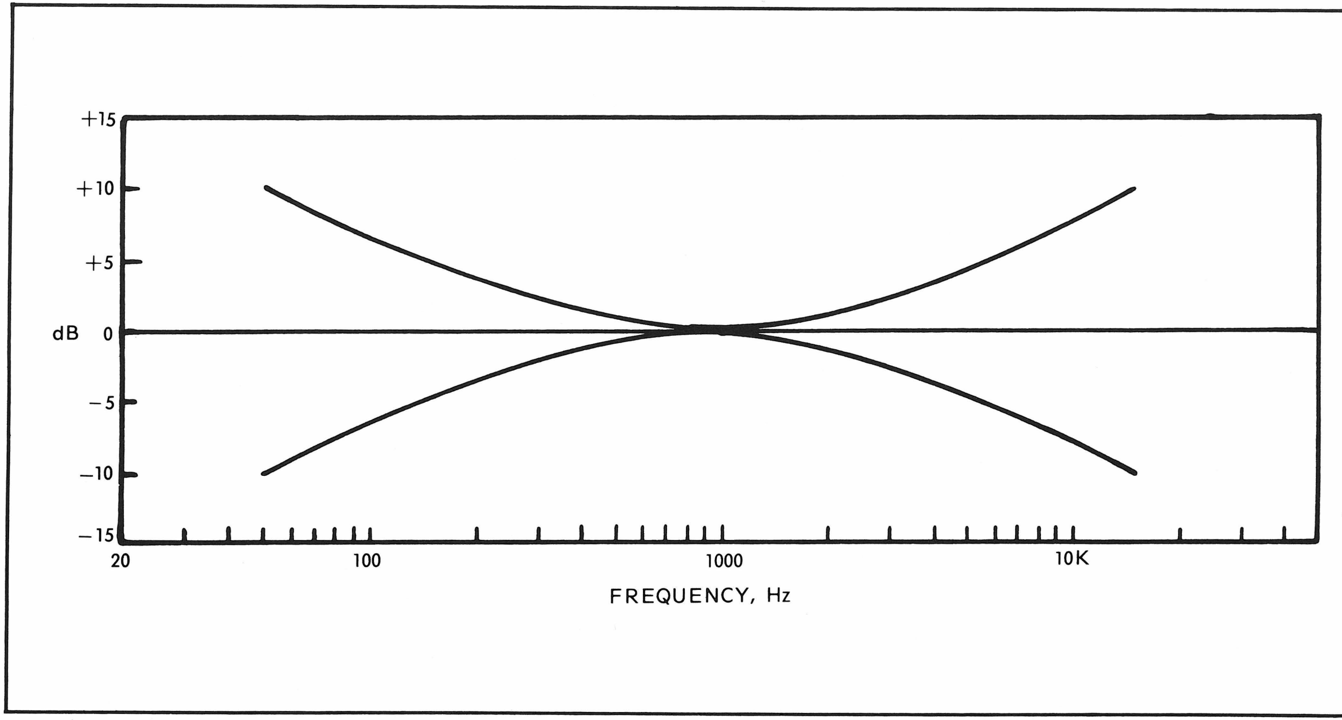

Fig. 14-10. The common tone control is very limited in adjustment of response.

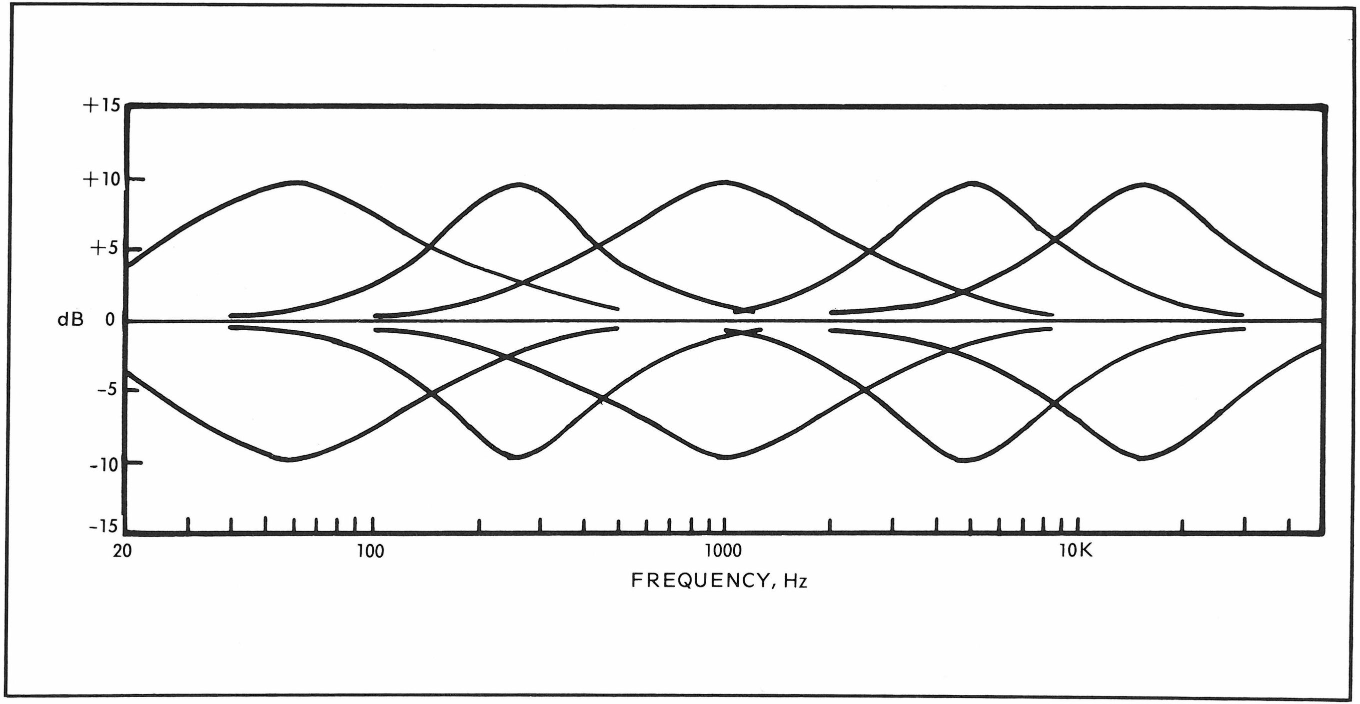

Fig. 14-11. A five band equalizer offers more flexibility than the common tone control, but still is very limited.

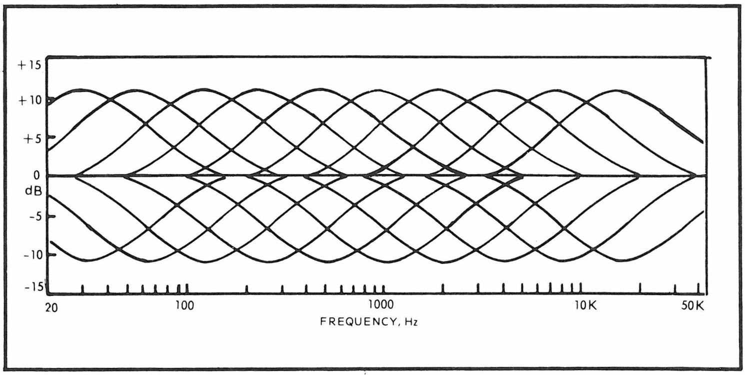

Fig. 14-12. The ten band equalizer offers a fairly wide range of equalization at octave intervals.

level meter, otherwise plus or minus 1 dB accuracy can be expected. Directions for evaluating the type of sound field existing in a given room are provided by Hall Engineering in their instruction manual.

The Hall ATS-401 Acoustical Test Set consists of a signal generator and sound level meter, fully calibrated, in a handsome, fitted case. Such a system provides accurate manual means for (1) adjusting graphic and parametric equalizers, (2) optimizing loudspeaker placement, (3) setting loudspeaker level controls, (4) selecting loudspeaker crossover components, (5) tuning ducts and ports in loudspeaker enclosures, and (6) calibrating microphones.

TONE CONTROLS

Every hi-fi enthusiast has done something in the direction of acoustical response equalization with the bass and treble tone controls on amplifiers and loudspeakers. The range of the simplest form of the traditional tone control may be something like that of Fig. 14-10. It is obvious that such broad control can do little to smooth squiggles such as those appearing in Figs. 14-6 and 14-7.

THE GRAPHIC EQUALIZER

Graphic equalizers owe their name to the fact that the positions of the array of control arms provide a representation of the response curve selected. These equalizers are well suited for general contouring of the response. As the number of filters in the set is increased the response of each filter is narrowed and the resolution increased as shown in Figs. 14-11, 12, and 13.

A good example of the professional room equalization filter set

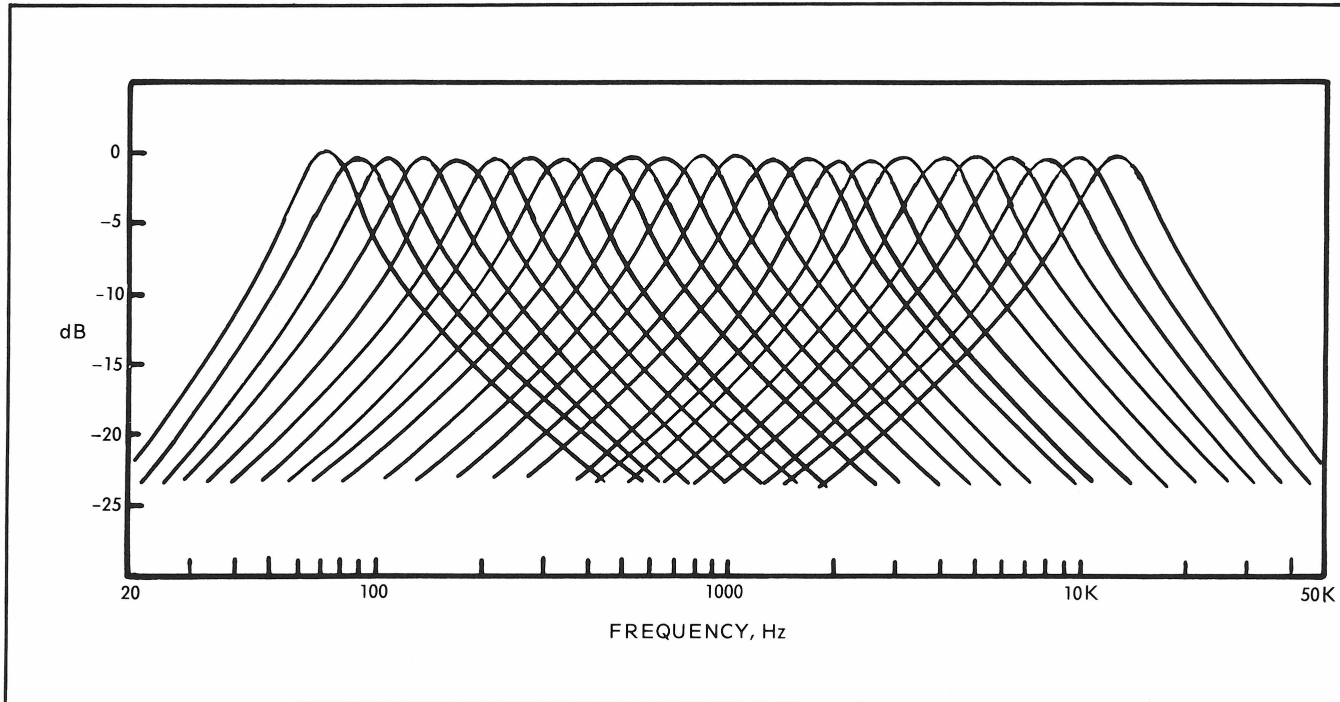

Fig. 14-13. Equalizers based on ⅓ octave intervals are flexible and widely used in smoothing acoustical response characteristics of listening rooms.

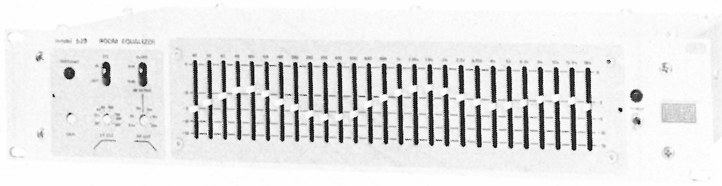

Fig. 14-14. The UREI Model 539 room equalization filter set. It contains 27 active filters which are centered on ISO one-third octave frequencies from 40 Hz to 16 kHz. These filters offer attenuation only, up to 15 dB, no boost provided. (United Recording Electronics Industries.)

is the UREI Model 539 shown in Fig. 14-14. It provides 0-15 dB of attenuation at each of its 27 frequencies. No boost is provided. The shape of a single filter set for maximum attenuation is shown in Fig. 14-15. This unit also provides a 12 dB per octave low cut filter continuously tunable from 20 to 250 Hz and a high cut filter tunable from 3.5 to 20 kHz.

The UREI dual graphic equalizer (Model 535) is shown in Fig. 14-16. Ten octave bands are provided in this instrument for each section, each capable of 12-dB boost or cut.

THE PARAMETRIC EQUALIZER

The parametric equalizer offers (1) adjustment of the frequency at which equalization peaks and dips are effective, (2) adjustment of the magnitude of the peak or dip, and (3) adjustment of the width of the peak or dip. Such flexibility is usually enhanced by the incorporation of several such units so that more than one irregularity of the curve can be smoothed.

An example of a dual parametric equalizer is the UREI Model 546 shown in Fig. 14-17. This equalizer provides four parametric filter sections in each channel. The extremes of bandwidth adjustments are shown in Fig. 14-18 to illustrate the extreme flexibility of the unit. Although only boost is shown in this figure, both boost and cut capabilities are present.

LIMITATIONS OF ROOM/LOUDSPEAKER EQUALIZATION

Why bother to acoustically treat the listening room? Why not correct for all its acoustical deficiencies by equalization? It sounds nice, but it will not work. Frequency response of the listening chain is the decisive factor in what the listener hears only if: (a) distortion, hum, and hiss are essentially absent in the loudspeaker output,

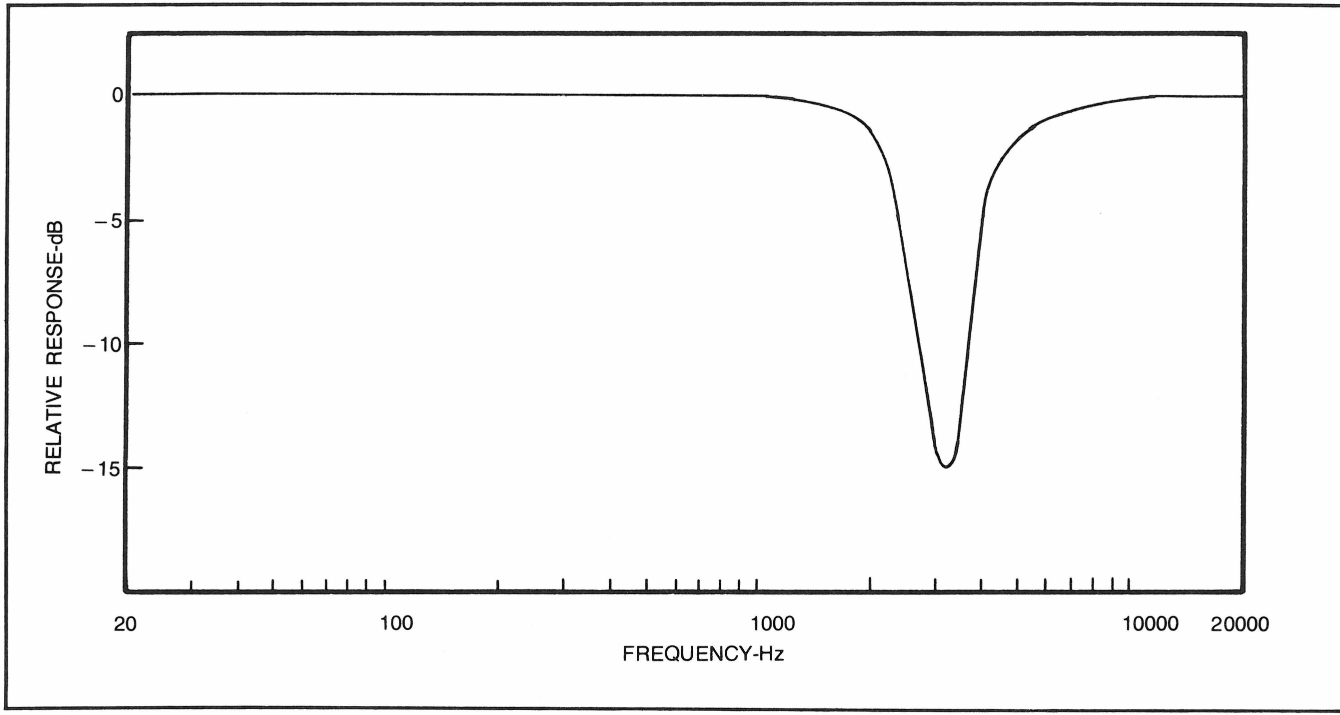

Fig. 14-15. The shape of the response of a single filter of the UREI Model 539 room equalization filter set adjusted for maximum attenuation.

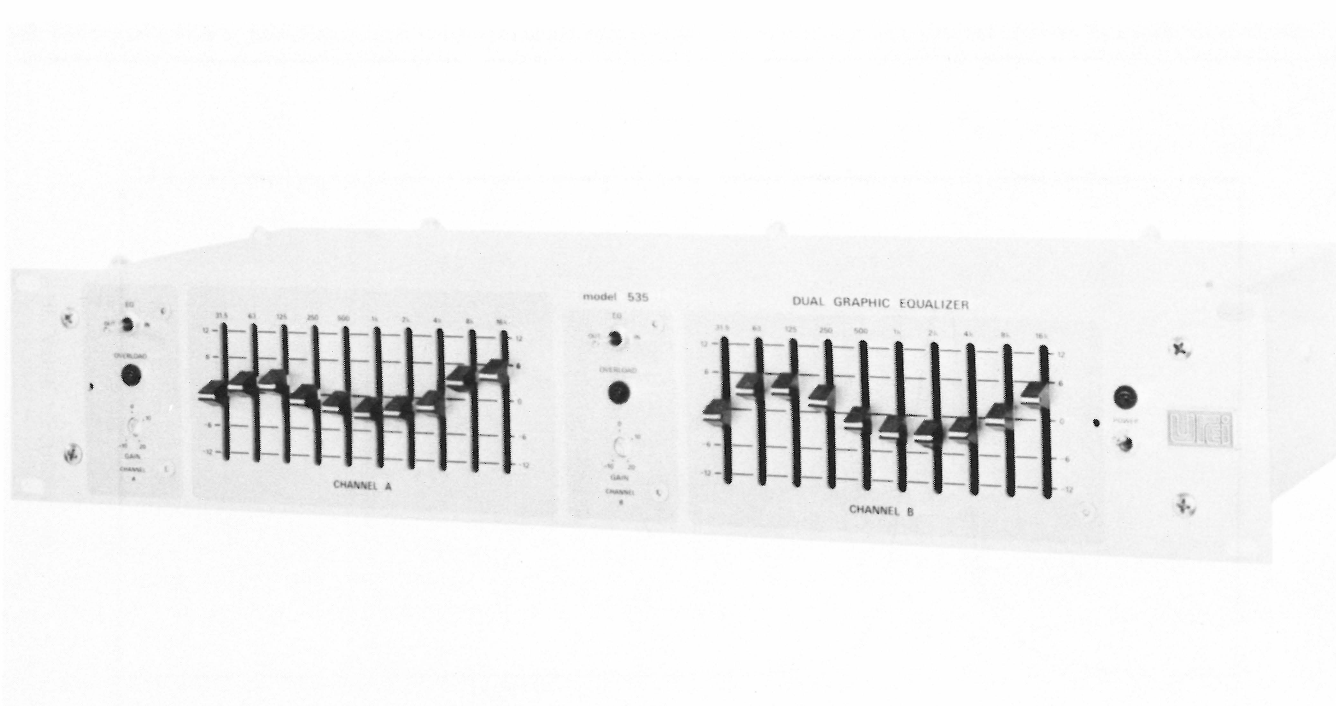

Fig. 14-16. The UREI Model 535 dual graphic equalizer. Each channel has 10 one-octave filter sections in each channel. (United Recording Electronics Industries.)

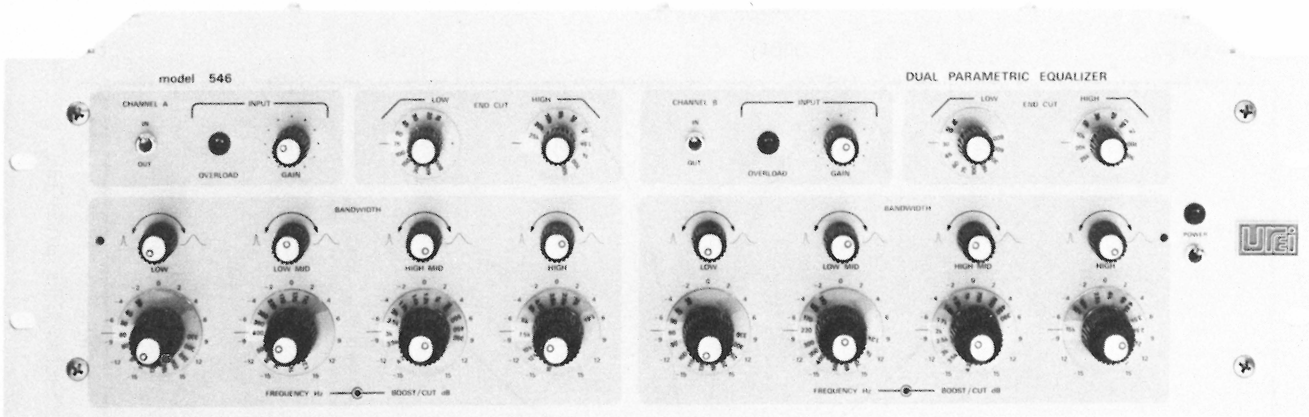

Fig. 14-17. The UREI Model 546 dual parametric equalizer. Four individual parametric units are incorporated in each channel, one for low band (30-330 Hz), one for low-midband (110 Hz - 1.2 kHz), one for high-midband (390 Hz-4.2 kHz), and a fourth for high band (1.4-15 kHz). Each unit may be set to any frequency in its band, its boost/cut may be adjusted up to 15 dB, and its bandwidth continuously adjustable ¼ to 4 octaves. (United Recording Electronics Industries.)

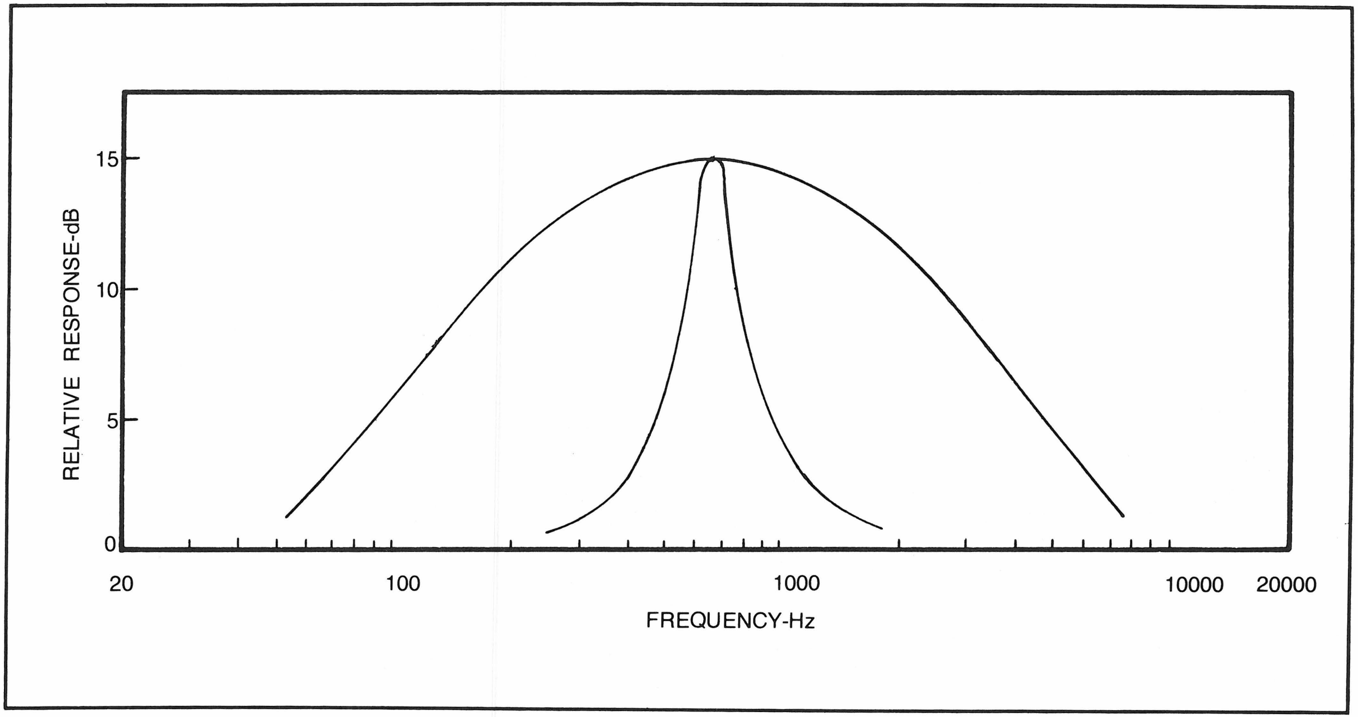

Fig. 14-18. The range of bandwidth adjustment in the UREI Model 546 dual parametric equalizer.

and (b) there are no gross acoustical defects in the listening room. Gross acoustical defects would include serious colorations due to normal modes, inadequate diffusion, improper reverberation time, or the presence of flutter. The process of room/loudspeaker equalization may reduce or eliminate colorations due to isolated modes or groups of modes, but the only safe way is to minimize such colorations before equalization.

Equalization works wonders in correcting those intangible, hard-to-get-at deficiencies of the room and the room/loudspeaker coupling but cannot cure every acoustical problem of the room or electro-acoustical problem of the loudspeaker.

The temptation to cure loudspeaker low frequency deficiencies by boosting response with an equalizer is greatly limited by amplifier power capability. For example, let us say that a 12 dB boost at 50 Hz would be desirable. A doubling of power represents only a 3 dB increase, hence 12 dB would represent 4 such doublings. Starting with a 100 watt amplifier, 200 watts are required for a 3 dB increase, 400 watts for 6 dB increase, 800 watts for 9 dB increase, and 1600 watts for the 12 dB desired. Such reasoning shows the fallacy of all but modest boosting at low frequencies.

It should also be emphasized that all such equalization as we have considered is for a steady state condition and that speech and music are highly transient in nature. When impulse response of equipment and rooms is considered, however, more sophisticated techniques and more complex measuring instruments are required. It is hoped that new developments in microprocessor controlled measuring instruments may make impulse response measurements commonly available and widely used.

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15

Evaluating Studio Acoustics

The early scientist was right when he said, "To measure is to know". Only by objective measurements may subjective judgments be kept within bounds. However, the very act of hearing is subjective and the ear must ultimately determine the acoustical quality of a space or a device. Objective measurements and subjective judgments, ideally, should go hand in hand, each serving as a control on the other.

By careful organization and planning, subjective (psychoacoustical) measurements are not only possible, but are increasingly being used.[1] By presenting samples of recorded sound in a statistically proper way to panels of listeners and asking for judgments on certain aspects of the sound, it is possible to obtain significant information out of the reach of physical instruments. The variability accompanying any measurements involving human judgment is a problem that requires that such tests be designed, conducted, and later analyzed in the right manner, preferably by experts in the field.

The task of evaluating any space to be used for recording or reproduction of sound boils down to determining the response of the room to various types of sound. In Chapters 5 and 6 the "sound" of a room was found to be determined by the normal modes. These normal modes dominate the low frequency portion of the spectrum in quite a complicated manner while at the higher frequencies the

normal mode density is so high that the ray concept can be applied. Therefore, the problem is to determine how the sound energy contained by the room surfaces responds to steady state and to transient signals.

STEADY STATE RESPONSE

The frequency response of an amplifier may be determined by injecting sine wave signals from an oscillator into the input and measuring the output at different frequencies. The resulting response graph is usually quite smooth and meaningful. Applying the same technique to a room is a "horse of a different wheelbase". A room has no specific input and output terminals, but the roomful of air can be energized (driven, excited) by a loudspeaker, and a microphone elsewhere in the room can give some idea of what happens between "input" point A and "output" point B. The result of such a test in a highly respected recording studio of 12,000 cu ft volume is shown in Fig. 15-1. The "input" signal was from a signal generator with automatic sine sweep. The signal picked up by the

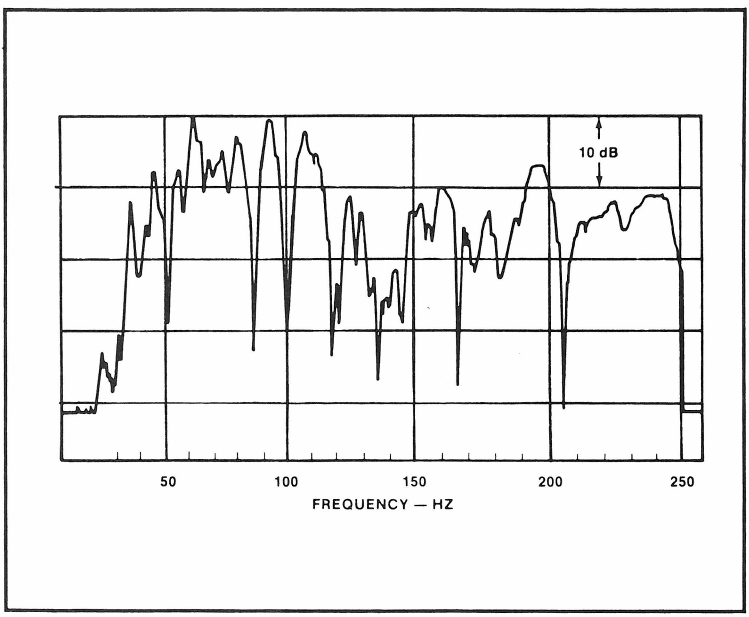

Fig. 15-1. Swept sine steady state response of a 12,000 cu ft recording studio. The loudspeaker was placed in a lower corner and the microphone in the diagonal upper corner of the room to assure that all room modes were energized and their effect recorded. Fluctuations of this magnitude are found in all studios and control rooms at the low frequency end of the audio spectrum.

microphone was recorded on ¼″ tape and later played into a Brüel & Kjaer Model 2305 graphic level recorder. The loudspeaker was placed in one corner and the microphone in the diagonal, opposite corner to couple to all room modes. The effect of the loudspeaker characteristic, included in Fig. 15-1, is negligible above 150 Hz, but the broad peak between 50 and 125 Hz is due to the increased radiation efficiency resulting from the corner position. No filters are used in this test at all; aside from the broad loudspeaker peak all fluctuations are due to the varying modal resonance support offered the energizing sine waves.

Fifty years ago considerable talent was expended in analyzing such steady state room response curves as that of Fig. 15-1. Today there is agreement that the value of such tests is very limited. One exception is that such a tracing does give some information on the diffusion of sound in the room: the less the fluctuation, the better the diffusion. Aside from this we shall appreciate the complicated modal pattern better by studying Fig. 15-1, but then moving on to other room measurements.

ACOUSTICAL RESPONSE

If instead of sine waves a swept ⅓ octave band of pink noise were used in the above room response measurement, we would find the response of Fig. 15-1 much smoother. In fact, apart from location of the loudspeaker and microphone, this would make the test comparable to the acoustical response measurements of Chapter 14, using swept ⅓ octave pink noise instead of successive octaves of noise from the test disc. This is a valuable test as it involves the loudspeaker characteristics as it interfaces the room acoustics.

TRANSIENT MEASUREMENTS

When transients are mentioned, impulses come to mind. The impulse response of an audio room is quite revealing. The cathode ray oscillogram of Fig. 15-2 was taken in a small (too small) control room. A special impulse source was fired at the face of one of the loudspeakers and the measuring microphone was placed at the operator ear position. The impulse source was an air pistol which ruptures a thin paper disc when fired, giving an intense (144 dB sound pressure level at 1 meter) but narrow (less than 1 millisecond) pulse.[2] The measuring microphone was the ½″ condenser microphone of the Brüel & Kjaer Model 2215 sound level meter on an extension cable with the sound level meter itself serving as a

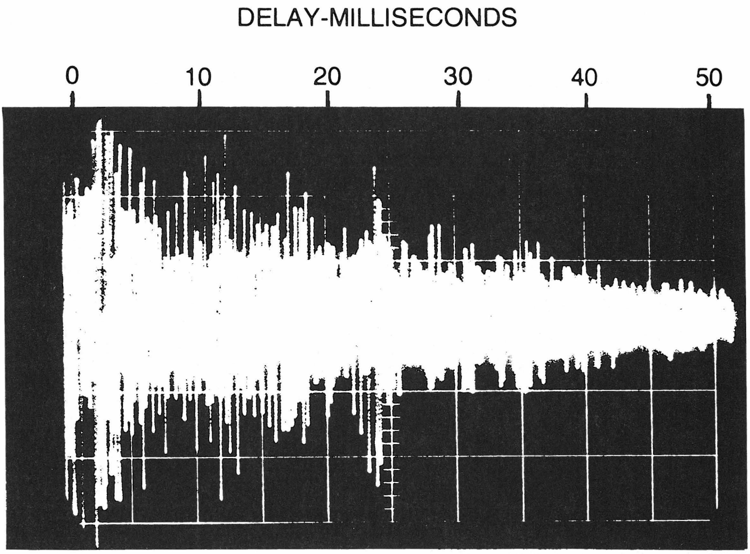

Fig. 15-2. Echogram recorded in a small control room revealing many problems. The impulse source was at the face of one of the loudspeakers and the direct pulse and the following torrent of reflected pulses were picked up by a microphone at the operator's ear position.

preamplifier. The received impulses were recorded on tape and later photographed from the screen of a Tektronix Model 515A trigger oscilloscope.

Although the echogram of Fig. 15-2 is actually a record of the first 50 milliseconds of the reverberatory decay, our interest is in the detail. The spikes represent discrete reflections from surfaces surrounding the loudspeaker, the mixing console, and various pieces of equipment surrounding the console. The number and amplitude of "echoes" arriving at the operator's ear position within the first 10 to 15 ms reveal that comparable early reflections picked up by the microphone in the studio will be completely masked by those in the control room. The Live End-Dead End™ control room is arranged so that these early reflections in the 10-15 ms region are of low amplitude so that early reflections in the studio can be heard in the control room.[3]

The reflections arriving at the operator's ear position within about 5 ms of the direct ray are also producers of comb filter distortions of the type described in Fig. 1-19. All of the reflection spikes of Fig. 15-2 produce comb filter effects, but those of the shorter delays create audible effects in the form of colorations of speech or music signals. In brief, the echogram depicted in Fig.

15-2 reveals serious problems in this control room which are the result of the basic design of the room (or the lack of it).

THE IMPULSER

Another application of impulses in the studio and control room is exemplified by the IMPulser device shown in Fig. 15-3 marketed by Acoustilog.[4] This is a relatively inexpensive black box which emits a positive-going pulse (similar to a $\cos^2$ pulse) with a frequency variable from 40 Hz to 10 kHz and a repetition rate from 0.3 to 10 pulses per second. The signal is fed to the amplifier driving the loudspeaker and the acoustical pulses are picked up by a microphone, amplified and displayed on any oscilloscope equipped with triggering facilities. The type of pulses shown in Fig. 15-4 then reveal significant information about the system. For example, the polarity of the left loudspeaker can be checked against that of the right. The polarity of the woofer can be checked against that of the midband radiator and the tweeter in a given loudspeaker. The time of arrival of signals from the different radiators in a given loudspeaker, or from different radiators in a given loudspeaker, or from different loudspeakers can be observed. Certain acoustical defects and effects of the room can be studied such as flutter echo between two parallel surfaces, rear wall slap echo, etc. Transmission loss measurements can be made with such impulses. High sound pressure levels can be generated non-destructively due to

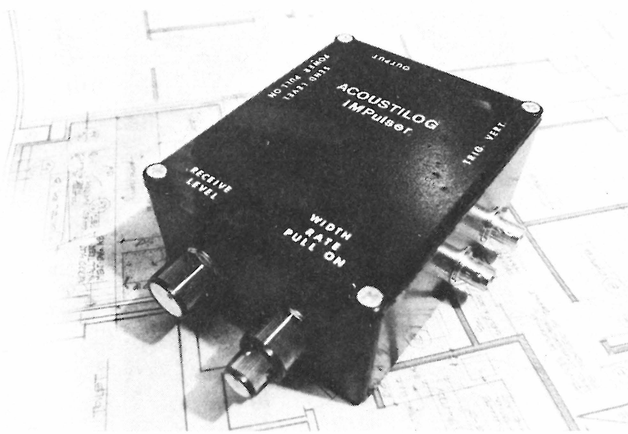

Fig. 15-3. The IMPulser, a generator of positive-going impulses of frequency variable from 40 Hz to 10 kHz and repetition rate from 0.3 to 10 pulses per second. Fed to an amplifier and loudspeaker and picked up by a microphone feeding a triggered oscilloscope, much information concerning polarity, time delay, and room acoustics is made available for study. (Acoustilog photo[3].)

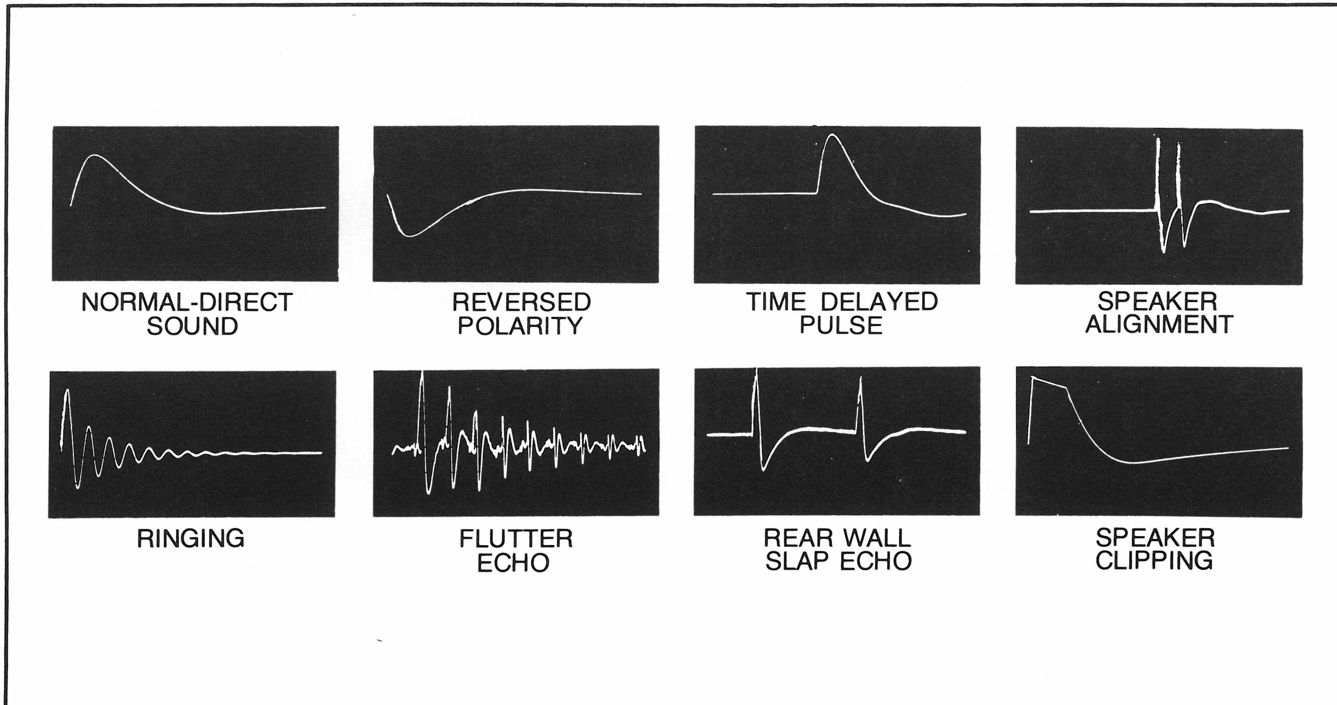

Fig. 15-4. Typical signals on a triggered oscilloscope resulting from use of the IMPulser. (Acoustilog photo[4].)

the low duty cycle of the pulses. The pulses can simulate a bass drum for exploration of isolating wall effectiveness, locating rattles, etc. It is nothing short of amazing what a simple impulse, adjustable in amplitude, duration, and repetition rate, can reveal about the functioning of electronic equipment, transducers, and room acoustics.

REVERBERATION TIME

Although the measurement of reverberation time is not everything, as inferred by some, it is an excellent beginning in understanding the acoustics of a space. A block diagram of equipment suitable for measuring reverberation time in the customary manner is shown in Fig. 15-5. The loudspeaker, driven by the amplified signal from the signal source, fills the room with sound as switch S is closed. The amplified signal picked up by the nondirectional microphone is fed either directly to a graphic level recorder for immediate measurement, or to a tape recorder for later measurement. Out of respect for the considerable weight of the graphic level recorder, the use of a tape recorder for data storage and later replay

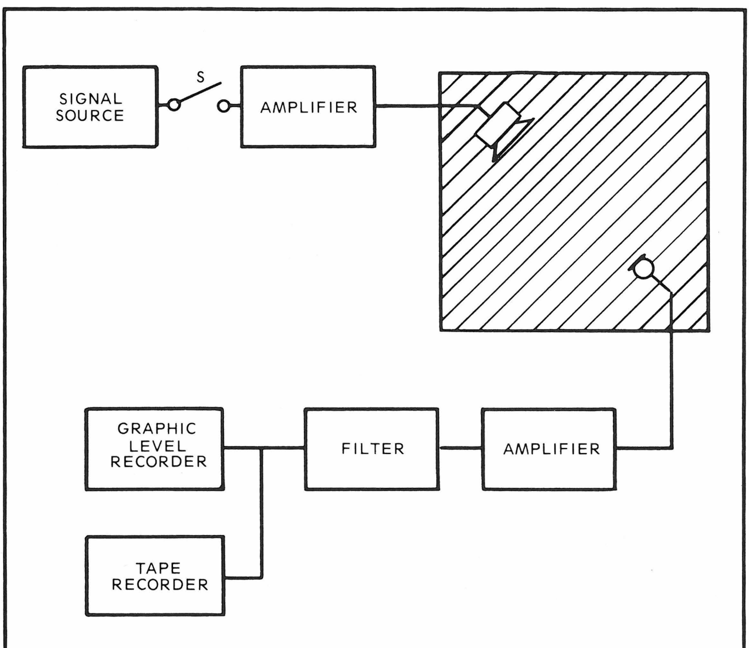

Fig. 15-5. The time-honored manner of making reverberation measurements in an enclosure.

Table 15-1. Organ as Sound Source.

Note	Frequency of note, Hz	Nominal Measuring Frequency, Hz
B_1	61.735	63
B_2	123.47	125
B_3	246.94	250
(Middle C)	261.63	----
B_4	493.88	500
B_5	987.77	1k
B_6	1,975.5	2k
B_7	3,951.1	4k

is far more popular. As the switch S is opened, the all-revealing decay of sound in the room is the significant event.

Various signals are used for room decay measurements, including pure tones, warbled tones, and random noise. Wallace Sabine, the Harvard acoustician who pioneered reverberation measurements at the turn of the century, used organ pipes as his signal source. In fact, the author recently recalled Sabine's method while visiting a recording studio in which the reverberation time was in question. The presence of a beautiful organ, complete with organist, and a control room full of recording gear suggested a repeat of Sabine's work to obtain rough data on existing reverberatory conditions. The organist was instructed to depress the keys of Table 15-1, suddenly releasing each one before depressing the next one while everything was recorded on tape. Subsequently playing the tape into the graphic level recorder gave a single decay at each frequency and a very rough fix on reverberation characteristics of the room. This sound source is not recommended for serious measurements, but in this case, a throwback to Sabine, initial data were obtained as a basis for later work.

Why are sine waves a troublesome signal for reverberation decay measurements? Figure 15-6 gives us a glimpse of the reasons for this situation. In this figure are reproduced the modal frequencies of the 23.3 × 16 × 10 Ft room from Table 6-3 and Fig. 6-4. The modes in the frequency region from 50 to 70 Hz are reproduced in Fig. 15-6. Only the one (0,0,1) axial mode at 56.43 Hz actually falls in this region, but the skirts of the 2,0,0 and the 0,2,0 mode do also. If a sine wave at 63 Hz is used to excite the room for a reverberation decay measurement, we see that it hits no modal frequency directly, but the skirts of some five modes are intercepted at different levels. These five modes **could** be excited in varying degrees by the 63 Hz signal, none to the fullest extent. These are forced

excitations. So much for modal excitation, what happens when the switch is opened? Each of these five modes decays from its respective original level of excitation and, note this, they decay at their natural frequencies, not 63 Hz. The recording of this decay will be a mixture of 70.54, 56.43, 66.55,59.86, and 61.39 beating together to yield a decay with wiggles determined by the different frequencies, the dominant ones tending to control the decay. If the room were excited by a ⅓ octave band of pink noise, all of the modal frequencies of Fig. 15-6 would be more fully energized and the decay would be somewhat smoother, but residual beating together of modal frequencies would still be evident. At 125 Hz, more modes would be excited by a ⅓ octave band of pink noise and the decay would be relatively smoother. At 250 Hz and higher, each band would embrace a consecutively greater number of modes and yield progressively smoother decays. Several types of decays are illustrated in Fig. 15-7. Figure 15-7(A) shows a typically smooth decay charac-

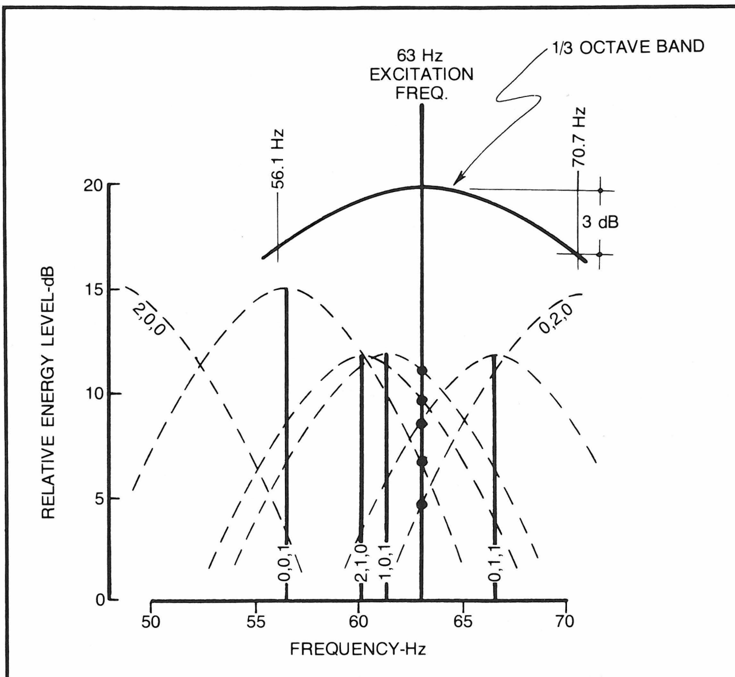

Fig. 15-6. The modal frequencies pictured are those of the 23.3 × 16 × 10 ft room of Table 6-3 and Fig. 6-4. The excitation of certain of these modes by a 63 Hz sine wave is illustrated as well as by ⅓ octave of pink noise. With either type of excitation there is a beating between the modes as they decay at their free frequencies.

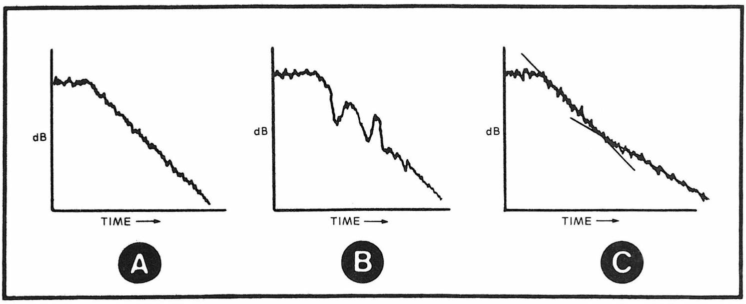

Fig. 15-7. Typical traces of the decay of sound in a room: (A) normal decay of well diffused sound resulting from inclusion of many modes in the band, (B) beating between modes during decay results in irregularity of the decay trace, and (C) a double slope decay may indicate the presence of a flutter echo.

teristic of bands of noise at higher frequencies and well-diffused conditions. The "wingdings" of Fig. 15-7(B) are typical of the beating together of adjacent modes decaying at their natural frequencies. Flutter echo may be the cause of a double-slope decay such as shown in Fig. 15-7(C).

The length of the decay (dynamic range) is influenced by the intensity of the original signal at the high end and the level of background noise at the low end as shown in Fig. 15-8. If the room is not excited to a high level and high noise prevails, the decay is short as in Fig. 15-8(A) and inaccurate determinations of the slope result. If the room is excited to a high level and low noise prevails, a long decay and accurate slope determination results as shown in Fig. 15-8(C). The capacity of the power amplifier/loudspeaker combination determines the level of excitation of the room and sound pressure levels of 100 to 110 dB are recommended as well as ear protectors. The effects of noise levels can be improved by proper

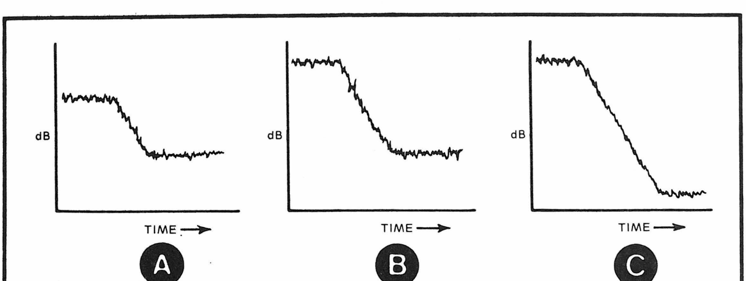

Fig. 15-8. The effect on decay traces of the degree of excitation of the room and the noise level: (A) low initial sound level and high noise level, (B) high initial sound level and high noise level, and (C) high initial sound level and low noise level.

filtering. For example, ⅓ or 1/1 octave filters can be used in recording the decays on tape and filters can also be used in the playback to the graphic level recorder. This double filtering can extend the dynamic range and improve the accuracy of decay slope measurement.

REVERBERATION TIME—A SPECIFIC CASE

The principles considered above were put into practice in the "as found" measurements in a 12,000 cu ft recording studio plagued with certain problems. Five successive decays of the octave band of pink noise centered on 125 Hz are shown in Fig. 15-9. The fluctuations are typical and it should be noted that there are major differences in the fluctuations from one decay to the next. The room remains the same, the equipment and technique are identical, why the difference? The one thing constantly changing is the random noise signal itself. In random noise the frequency, phase, and amplitude are constantly changing and the exact signal conditions prevailing at the instant the switch is opened to initiate a decay can be anything within a certain range. Just which modes within that band are excited, and to what degree, is constantly changing.

The variations in successive decays due to changes in random

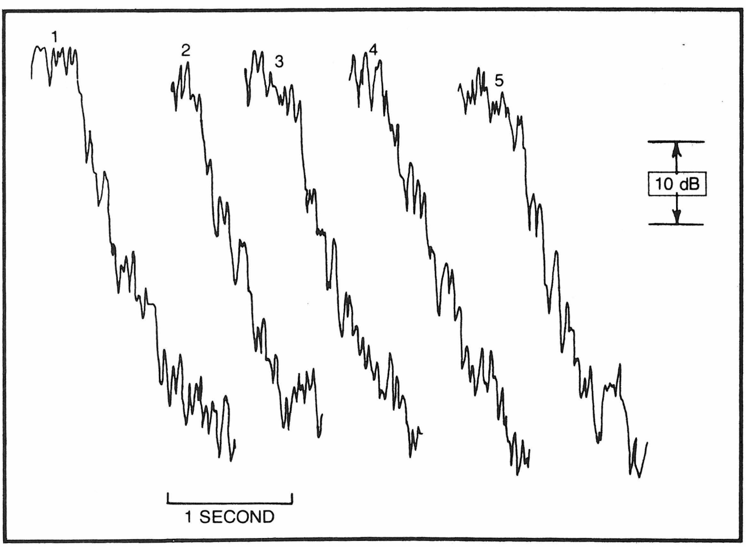

Fig. 15-9. Five successive decays of 125 Hz octave bands of pink noise in a studio of 12,000 cu ft volume. At the instant the switch is opened to initiate the decay the random noise energizing signal conditions are different each time resulting in the energizing of the modes in different ways.

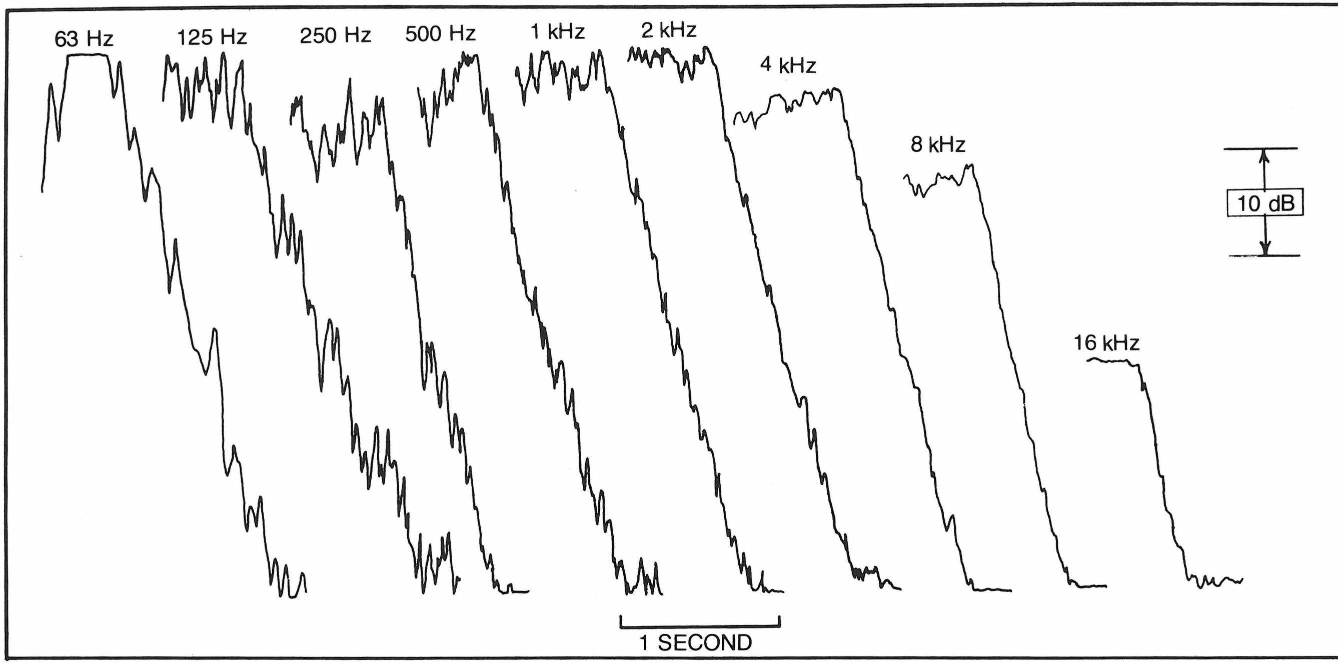

Fig. 15-10. Typical decays of octave bands of pink noise for a 12,000 cu ft studio. The 125 Hz decay is the #4 decay of Fig. 15-8. Notice that the more modal frequencies embraced in an octave, the smoother the decay. The shorter decays at 8 and 16 kHz are due principally to reduce output of loudspeaker and response of tape recorder at these frequencies, but there is still sufficient trace to determine the slope.

noise are matched or exceeded by changes from octave band to octave band as illustrated by the selected traces of Fig. 15-10. The higher the band center frequency, the greater the mode density and the smoother the trace.

Table 15-2 lists the reverberation times calculated from decay slopes for 9 frequencies, 4 microphone positions, and 5 decays for each frequency/microphone condition. From this tabulation something of the magnitude of experimental variations can be sensed. The mean (average) values of reverberation time, lumping data from all four microphone positions together, are listed at the bottom of the page. The standard deviation for each mean value is also given which is a statistical measure of the data spread. Assuming a gaussian (normal) distribution of the data points, the mean value plus and minus the standard deviation encompasses 67% of the data points. For example, for the 500 Hz frequency the mean reverbera-

Table 15-2. Reverberation Time Data Sheet.

MIC	Octave Band Center Frequency—Hz								
POS.	63	125	250	500	1k	2k	4k	8k	16k
M-1	0.63	0.80	0.48	0.54	0.49	0.56	0.44	0.45	0.32
	0.64	0.66	0.43	0.57	0.56	0.47	0.47	0.43	0.33
	0.66	0.80	0.40	0.54	0.57	0.56	0.50	0.43	0.32
	0.69	0.88	0.40	0.61	0.56	0.56	0.50	0.44	0.36
	0.56	0.61	0.46	0.58	0.66	0.54	0.46	0.43	0.34
AVE →	0.64	0.75	0.43	0.57	0.57	0.54	0.47	0.44	0.33
M-2	0.53	0.63	0.41	0.48	0.54	0.59	0.51	0.44	0.40
	0.70	0.59	0.47	0.50	0.54	0.56	0.50	0.40	0.40
	0.68	0.55	0.48	0.55	0.53	0.55	0.54	0.41	0.34
	0.62	0.67	0.50	0.50	0.54	0.56	0.51	0.36	0.35
	0.71	0.61	0.40	0.49	0.57	0.56	0.48	0.42	0.40
AVE →	0.65	0.61	0.45	0.50	0.54	0.56	0.51	0.41	0.38
M-3	0.76	0.72	0.41	0.55	0.56	0.52	0.49	0.38	0.38
	0.74	0.55	0.35	0.63	0.55	0.56	0.48	0.37	0.34
	0.73	0.65	0.33	0.54	0.59	0.52	0.48	0.39	0.34
	0.68	0.68	0.33	0.50	0.56	0.53	0.47	0.37	0.32
AVE →	0.80	0.58	0.38	0.57	0.58	0.56	0.49	0.38	0.34
	0.74	0.64	0.36	0.56	0.57	0.54	0.48	0.38	0.34
M-4	0.63	0.68	0.44	0.55	0.62	0.56	0.48	0.41	0.38
	0.70	0.60	0.46	0.56	0.63	0.56	0.52	0.40	0.41
	0.71	0.56	0.40	0.55	0.59	0.57	0.52	0.40	0.37
	0.68	0.63	0.40	0.58	0.59	0.60	0.52	0.42	0.36
AVE →	0.62	0.67	0.43	0.50	0.58	0.56	0.51	0.41	0.34
	0.67	0.63	0.43	0.55	0.60	0.57	0.51	0.41	0.37
MEAN	0.67	0.66	0.42	0.54	0.57	0.55	0.49	0.41	0.36
STD. DEV.	0.06	0.09	0.05	0.04	0.04	0.03	0.02	0.03	0.03

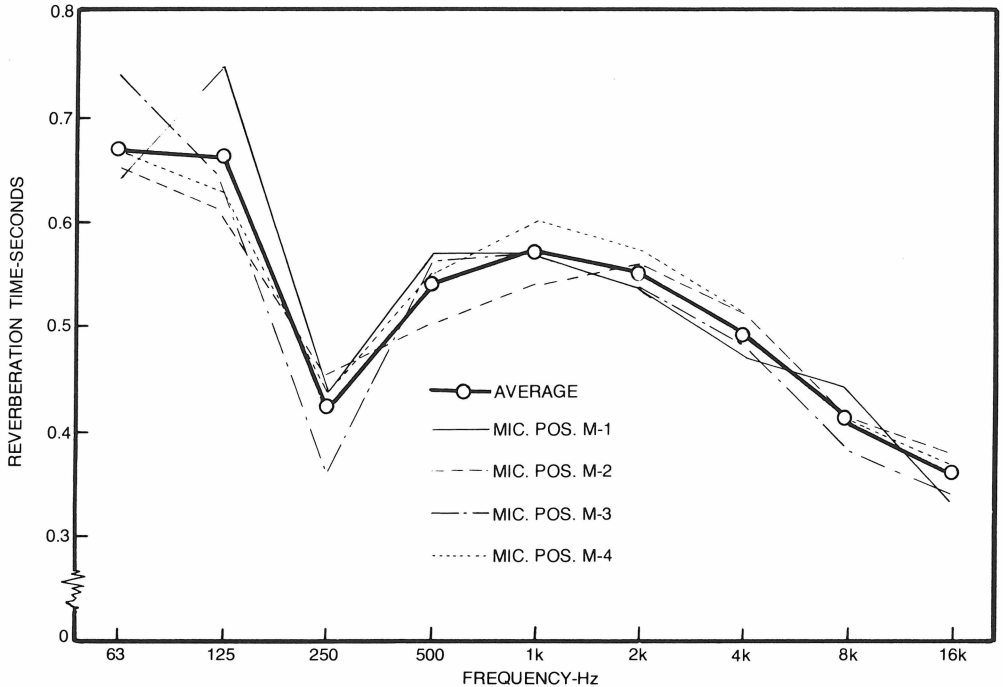

Fig. 15-11. The "as found" reverberatory conditions of a 12,000 cu ft studio. Reverberation time varies with position in the room because of standing wave conditions. The dip in the reverberation time at 250 Hz, measured at all positions, results from a resonant absorber on the walls.

tion time is 0.54 second with a standard deviation of 0.04 which means that 67% of the data points fall between 0.50 and 0.58 second. The modest standard deviations shown testify that significant mean values of reverberation time have been obtained.

The data of Table 15-2 are plotted in graphical form in Fig. 15-11. The averages of each of the four microphone positions are also plotted, showing good consistency in spite of variations from point to point in the room.

REVERBERATION TIME—THE EASY WAY

The method of measuring reverberation time outlined above is tedious and time consuming. The experienced consultant often prefers it in spite of the tedium because of the opportunity it gives to study the detail of each reverberatory decay as traced by the high speed graphic level recorder. The Acoustilog Model 232 Reverberation Timer pictured in Fig. 15-12 requires only the push of a button to determine a reverberation time. Reading the delay time on the LED digital readout is simplicity itself, but the detail of the decay is lost unless an expensive storage oscilloscope is lugged to the measurement site. However, the detail of the decay is not always all that important, and the convenience of one of the available reverberation timers such as the Model 232 cannot be denied.

The Acoustilog 232 computes reverberation time in each of seven octave bands. It has its own pink noise source built in. There are actually two complete sets of octave filters in the 232, one filtering the gated pink noise signal sent to the user-supplied power amplifier and loudspeaker and the other filteirng the received signal from the user-supplied microphone. Doubling the filtering in this way results in a net overall effective bandwidth of approximately ½ octave. The crystal controlled oscillator in the time base yields a resolution of 0.01 second (10 milliseconds) which means that the

Fig. 15-12. The Acoustilog Model 232 Reverberation Timer. Reverberation time may be measured in 7 bands from 125 Hz to 8 kHz or flat. An internal pink noise generator supplies room excitation signal to a user-supplied power amplifier and loudspeaker. The time for 20 or 30 dB decay is read out on a LED digital indicator. The reverberation time is then 3 times (for 20 dB) or 2 times (for 30 dB) the indicated delay time. (Acoustilog photo[4].)

reverberation time of small, dead rooms less than 0.3 second (300 milliseconds) can be readily measured.

The Model 232 reverberation timer measures the time between the interruption of the signal and either the −20 dB or −30 dB point selected by the user. As the definition of reverberation time is the time required for the sound pressure level to fall to −60 dB, the LED readout figure must be multiplied by 3 for the −20 dB setting or 2 for the −30 dB setting. The experts tell us that what happens during the first 30 dB of decay is what is important to the ear. The Model 232 reverberation timer is contained in only a 1¾″ high standard 19″ panel and weighs only 4 lbs. The IMPulser can be mounted in the Model 232 on special order.

It should be mentioned that frequency response of power amplifier, loudspeaker, and/or tape recorder (if used) are not very important in reverberation measurements. All that is being measured is the time rate of change of sound level which is unaffected by response irregularities as long as sufficient initial level is available.

NOISE SOURCES

Illumination fixtures are often a source of buzz or hum, often at twice the frequency of the power source. The filaments of incandescent lamps can generate audible as well as radio frequency interference. Fluorescent fixtures are a very common source of troublesome buzz, usually traceable to loose laminations in the ballast reactors. It is strongly urged that fluorescent ballast reactors be banished to a box well removed from the recording area.

Air conditioning equipment is notorious as a producer of noise. Chapter 8 points out that such sounds reach the studio either by an airborne or structure-borne path, or possibly a complex combination of both. Let us assume that a ventilating fan noise is giving trouble in a studio. How does one determine whether it comes via air or structure? If the whole studio shakes to the tune of the fan, it may be obvious that it is a building vibration problem requiring a resilient mounting for the machinery. On the other hand, the transmission path by which the sound reaches the studio might be more difficult to determine.

One way of solving such a problem is to record the sound close to the offending source and play this back on a loudspeaker placed in the equipment room but with the machinery silent. The level is adjusted to approximate that which the machinery would produce if it were running. Now we have a producer of the noise which is not fastened to the structure. If the noise radiated by the loudspeaker is

not heard in the studio, the problem should be attacked on a structure-borne basis. One should also be alert to the possibility of fan blade modulation noise being transmitted through the air supply and exhaust ducts. This can be reduced by lining the ducts with glass fiber board or the use of glass fiber ducts. Or recourse could be made to absorbent baffles in a noise trap, installation of plenums, or even tuned acoustic stubs.

EVALUATION OF BACKGROUND NOISE

The ear is the final arbiter in the evaluation of background noise. If extraneous noise can be heard on the final recorded product, the noise level certainly is too high and must be reduced. If studio noise the ear can detect is lost in the normal noises of the recording system, perhaps studio noise level is satisfactory. The proper time to guard against noise intrusion is during the design and construction stages and the Noise Criteria (NC) curves offer an excellent way of designing for acceptable final noise levels and evaluating the noise of existing structures. The family of Noise Criteria curves, shown in Fig. 15-13, make possible a single number description of noise spectra. Strictly speaking, the single number applies exactly only when the noise spectrum exactly matches the NC contour.

The background noise in recording studios, control rooms, and listening rooms usually falls within the range NC 15-25. Some top-flight rooms in very quiet areas approach the NC-10 contour which is not even shown in Fig. 15-13. The octave band noise spectra of a certain studio plotted on Fig. 15-13 would be classed as NC-24 because the peak of its spectrum reaches that contour. It would be considered barely acceptable for average work and unacceptable for very critical work.

By plotting on the NC family of contours the measured environmental noise to which a proposed studio would be exposed and also marking the NC goal contour desired within the studio when completed, the requirements of isolating walls, ceilings, and floors is then the difference between the two spectra.

FLUTTER ECHO

Highly reflective parallel walls tend to give rise to flutter echoes which often can be detected by the unaided ear. A clap of the hands provides an excellent impulsive sound for exciting a flutter mode. Before trying this in a relatively dead studio, it is well to

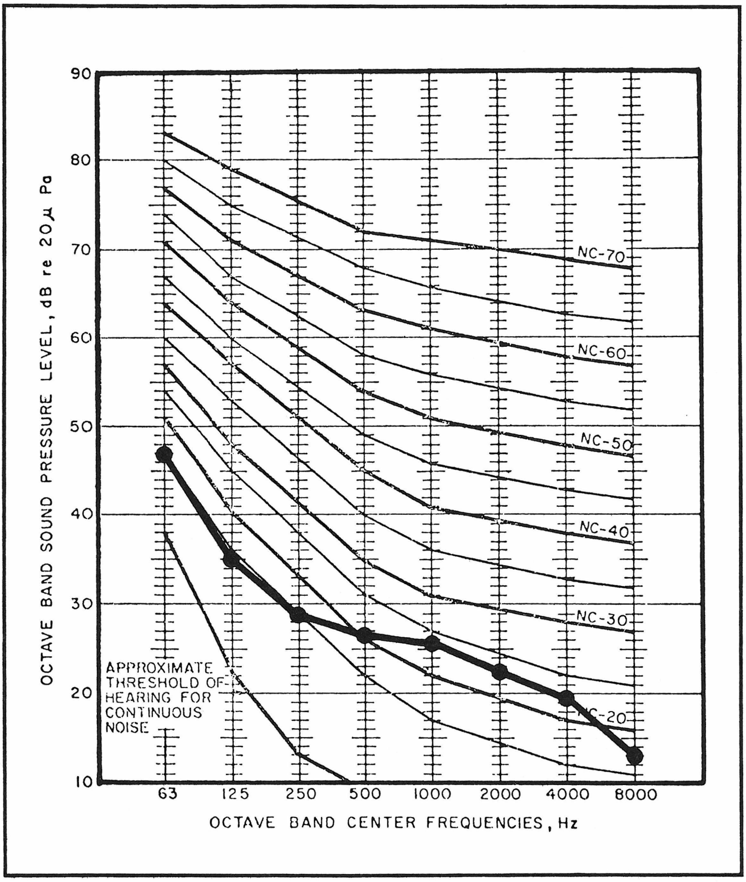

Fig. 15-13. Noise Criteria (NC) contours for evaluating measured noise spectra. The shape of these contours is determined by the sensitivity of the human ear.

become familiar with the procedure where the flutter is very prominent, such as a large room with parallel, hard surfaced walls. After each handclap a flutter decay should be clearly evident. With a bit of such practice, detecting any flutter which might exist in the studio should be considerably easier.

Since speech sounds are impulsive by nature, the effects of flutter degrade speech more than music. As we have seen, flutter tends to increase the reverberation time (Fig. 15-7C). This results from the fact that the surfaces involved in the flutter are less absorptive than those dominating diffuse sound. The presence of observable flutter will almost certainly indicate the presence of a serious low frequency resonance problem.

PINPOINTING ROOM COLORATIONS

Applying a good critical ear to male speech picked up from a room over a high quality system is one quite effective way to detect axial mode colorations. Only the more prominent colorations can be singled out by this method, however, and it is difficult to estimate the precise frequency at which they occur so that corrective measures may be taken.

The BBC used an interesting device to assist the ear, a tunable amplifier that amplifies a narrow frequency band (10 Hz) about 25 dB above the rest of the spectrum. A small portion of the output of this selective amplifier is mixed with the original signal so that its contribution is scarcely noticeable except when tuned to the frequency of a coloration. In this way the colorations are easily detected and their frequency determined. Using this instrument in many treated studios disclosed the fact that one or two colorations in a studio were quite common and that most of them are between 100 and 175 Hz with a smaller number around 250 Hz as seen in Fig. 6-14.

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2. Matsudaira, T. Ken, Tadashi Takise, Keisuke Sekiguchi, and Osamu Yamaguchi. *Fast Room Acoustic Analyzer (FRA) Using Public Telephone Line and Computer.* Jour. Audio Engr. Soc., Vol. 25, No. 3 (March 1977) pp. 82-94.
3. Davis, Don and Chips Davis. *The LEDE™ Concept for the Control of Acoustic and Psychoacoustic Parameters in Recording Control Rooms.* Jour. Audio Engr. Soc., Vol. 28, No. 9 (September 1980), pp. 585-595.
4. Acoustilog, 19 Mercer Street, New York, NY 10013.

16

A Pictorial Tour of Studios Around the World

This tour is conducted for only one reason, to catch a glimpse of features and ideas which might be of interest both to the high fidelity enthusiast and the studio operator. We are going to visit some midgets and some giants, but in none of them do we indulge in anything like comprehensive coverage. The amateur can learn from the professional and, who knows? Perhaps the professional can learn something from the little fellow away out in Timbuctu struggling along with little sympathy and even less money.

There is a prodigious amount of recording going on in the "third world," generally for Christian radio use, by a corps of people whose special training for the job often falls far short of their enthusiasm and dedication. They share with the average high fidelity enthusiast that gnawing compulsion to improve the quality of their work even though there is rarely money enough to go first class.

So, keep your eyes open. Maybe there are a few ideas in the following 45 photographs and supporting diagrams that will be worth the price of admission!

Fig. 16-1. Doors with sufficient transmission loss may be a problem if money is scarce. The Far East Broadcasting Company in Manila is very happy with the inexpensive doors they have developed over a period of years. In these general views the magnetic seals of the type commonly used on home refrigerator doors are seen as white lines. For the sides and the top, the seals are mounted to the door casing with the companion iron straps on the door. At the bottom the sealing strip is mounted on the door and the iron strap on the raised threshold to minimize damage from feet.

Fig. 16-2. Doors with sufficient transmission loss may be a problem if money is scarce. The Far East Broadcasting Company in Manila is very happy with the inexpensive doors they have developed over a period of years. In these general views the magnetic seals of the type commonly used on home refrigerator doors are seen as white lines. For the sides and the top, the seals are mounted to the door casing with the companion iron straps on the door. At the bottom the sealing strip is mounted on the door and the iron strap on the raise threshold to minimize damage from feet.

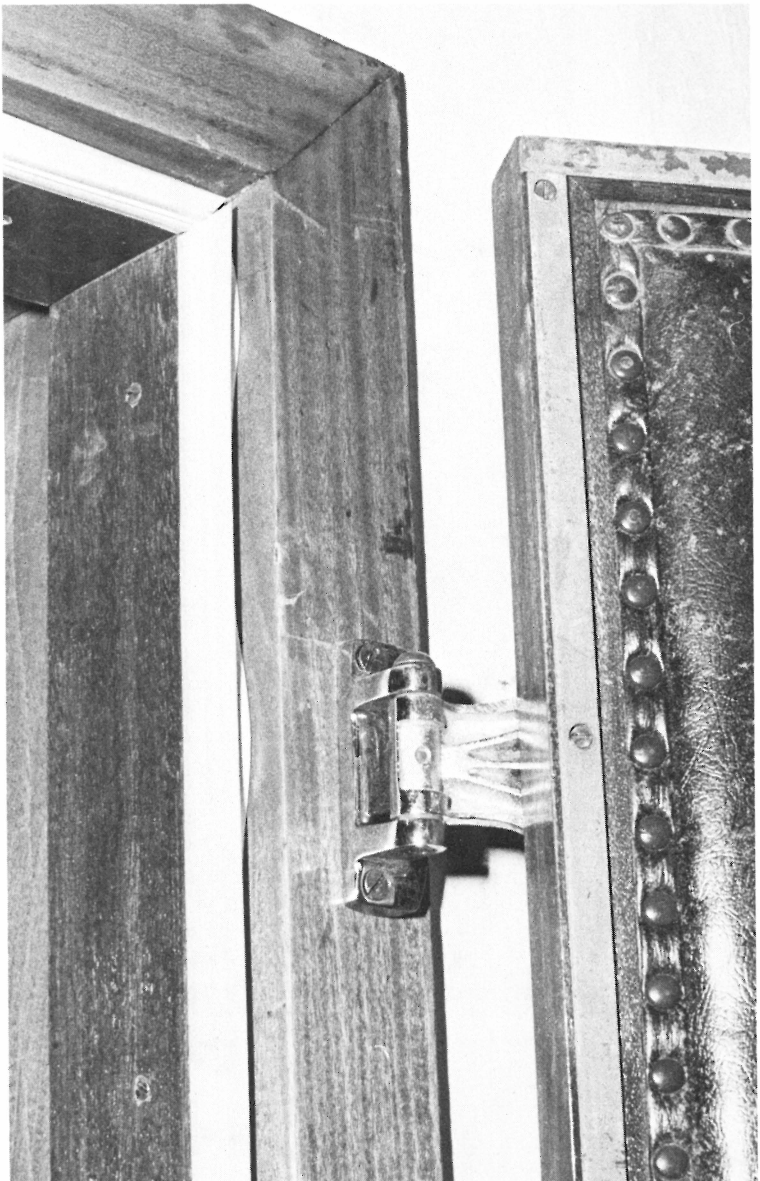


Fig. 16-3. Heavy freezer door hinges are utilized to swing the heavy doors. The magnetic sealing strips and iron straps (see Fig. 8-12) are clearly seen at top, side, and bottom edges of the door and frame.

Fig. 16-4. Heavy freezer door hinges are utilized to swing the heavy doors. The magnetic sealing strips and iron straps (see Fig. 8-12) are clearly seen at top, side, and bottom edges of the door and frame.

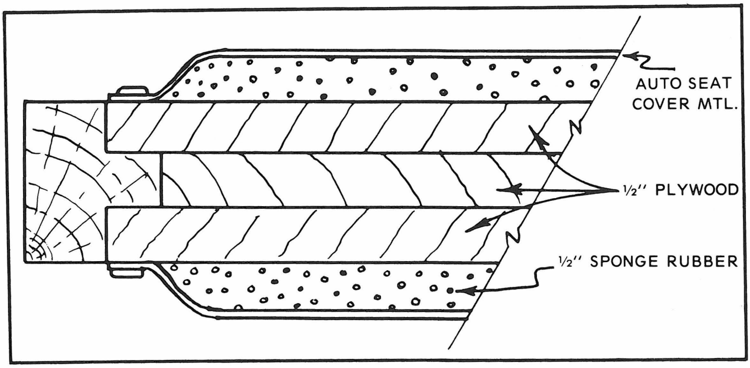


Fig. 16-5. The door is made of three sheets of ½″ plywood glued together and fitted with a mortised edging. Both sides are covered with ½″ sponge rubber which, in turn, is covered with automobile seat cover material. (Far East Broadcasting Company photos.)

Fig. 16-6. The studio built by CAVE (pronounced "Cah 'vay") of Centro-Audio-Visual-Evangelico in Campinas, S.P., Brazil was designed by Dr. William E. Haney, now of Far East Broadcasting Company. First we note that the polycylindrical diffusers are of assorted sizes, the larger ones being more effective as diffusers at the lower frequencies. The drapes absorb high frequency energy when closed and reveal a highly reflective wall surface of corrugated asbestos roofing material when retracted, giving some control of the treble reverberation time. (Moody Institute of Science photo.)

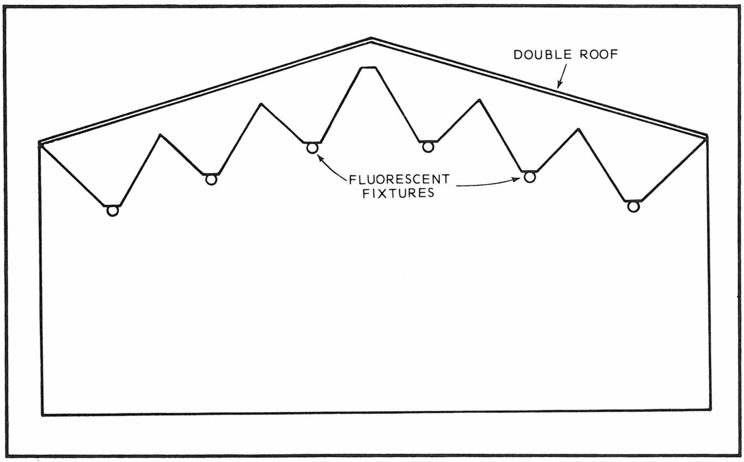


Fig. 16-7. The serrated ceiling in the CAVE studio effectively prevents flutter between it and the floor. The absorption of the ceiling is high because much of the sound undergoes several reflections between adjacent surfaces.

Fig. 16-8. The AVEC (Audio-Visual-Evangelism-Committee) studio of the Hong Kong Christian Council in Hong Kong. The studio was designed by Mr. Delbert Rice. Polycylindrical diffusers much shorter than wall height dominate the room. These provide needed low frequency absorption and their large chord dimension would suggest that they are effective to quite low frequencies. Some of these polys hide building support columns.

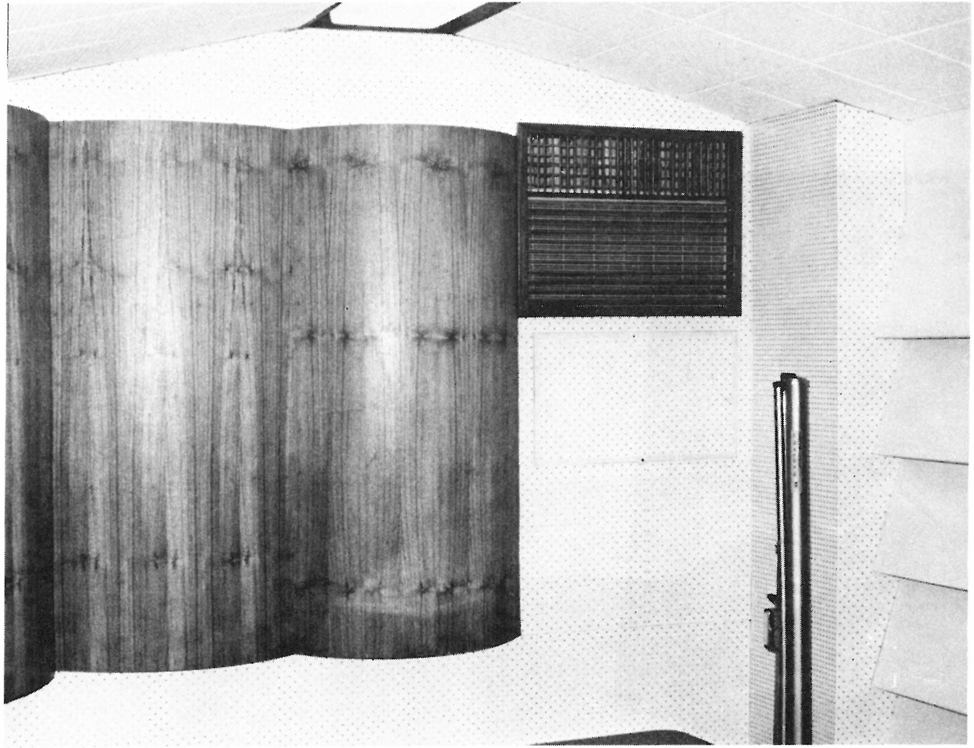


Fig. 16-9. The concave ceiling, designed to discourage floor/ceiling reflections, is probably not concave enough to cause focusing problems.

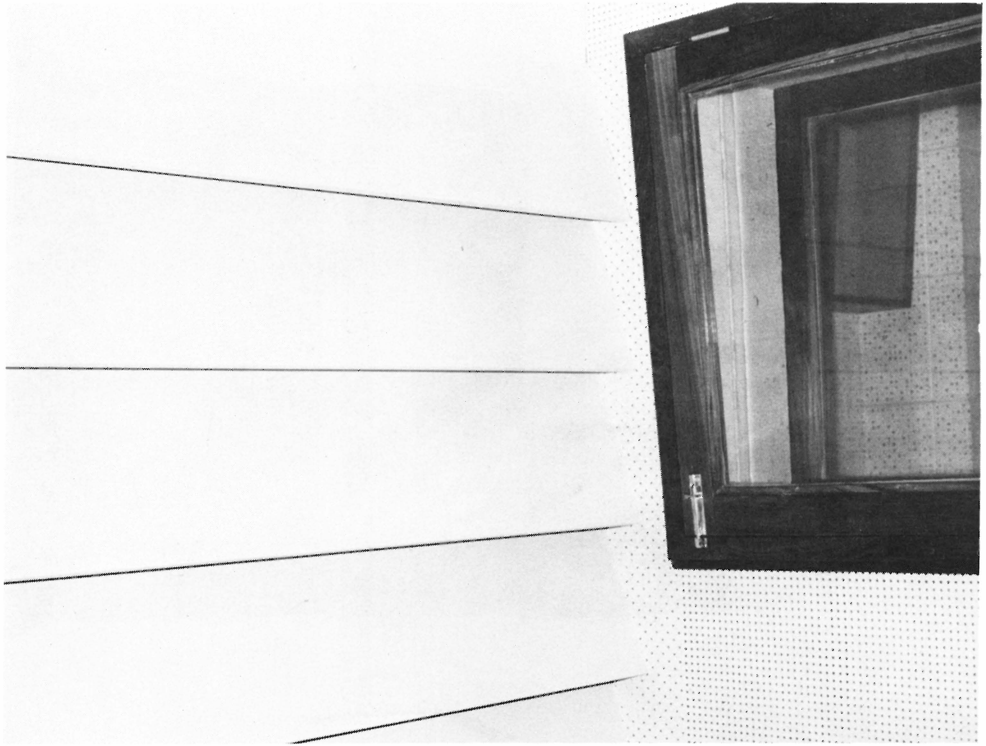


Fig. 16-10. This louvered structure which dominates one wall is reported to be built for low frequency absorption. It is not clear how this structure is supposed to accomplish this. It certainly could be expected to have an absorption and diffusing effect as described in Fig. 16-11.

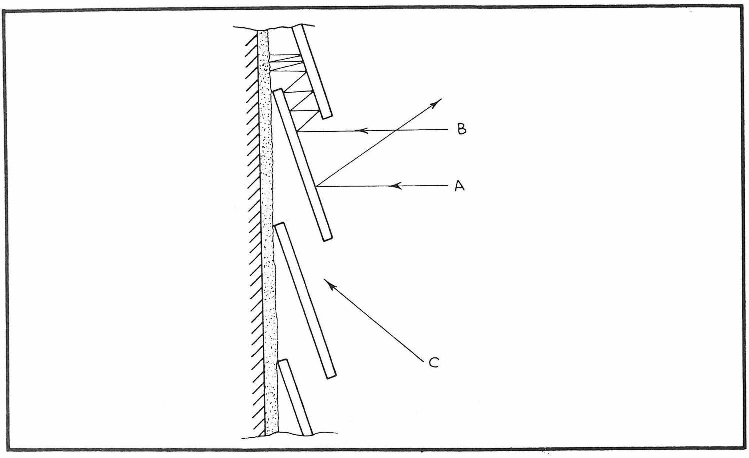


Fig. 16-11. The louvered structure of Fig. 16-10 provides diffusion at higher audio frequencies and acts as an acoustical "sink" or trap. Horizontal ray A is reflected from the wooden louver externally while ray B is subjected to multiple reflections within the trap. Reflections C from the floor would tend to be completely absorbed.

Fig. 16-12. A studio used primarily for language dubbing in Mexico City. The studios were designed by Mr. Michael Rettinger, well known acoustical consultant. The slat resonators have slits of different widths to broaden the peak of absorption. (Moody Institute of Science photo.)

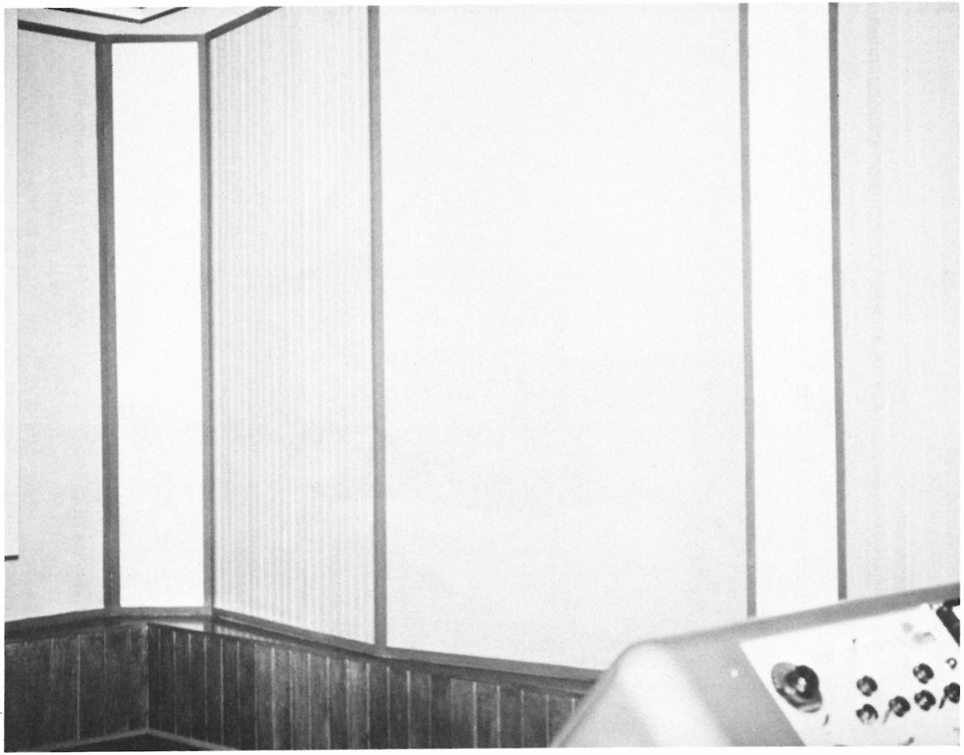


Fig. 16-13. The walls are of splayed panels. Some panels are slat resonators, some are perforated hardboard which covers glass fiber, and some are highly reflective surfaces. Both the angular surfaces and the distribution of different panels contribute to sound diffusion in the room.

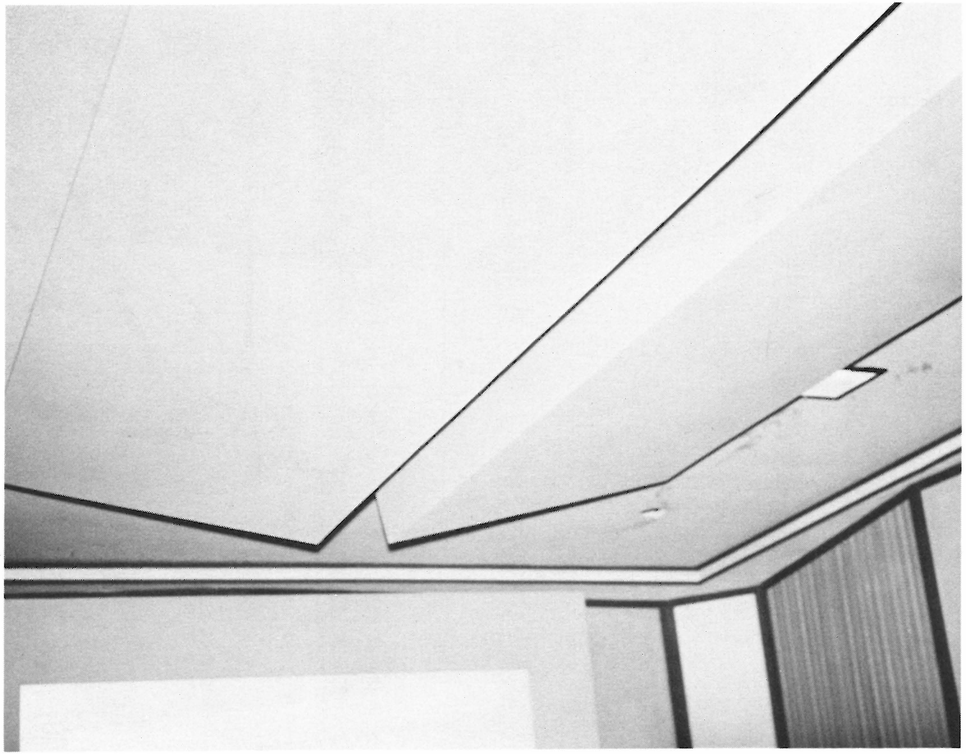


Fig. 16-14. A simple plywood-faced geometrical ceiling feature supplies some low frequency absorption and breaks up floor-ceiling reflections.

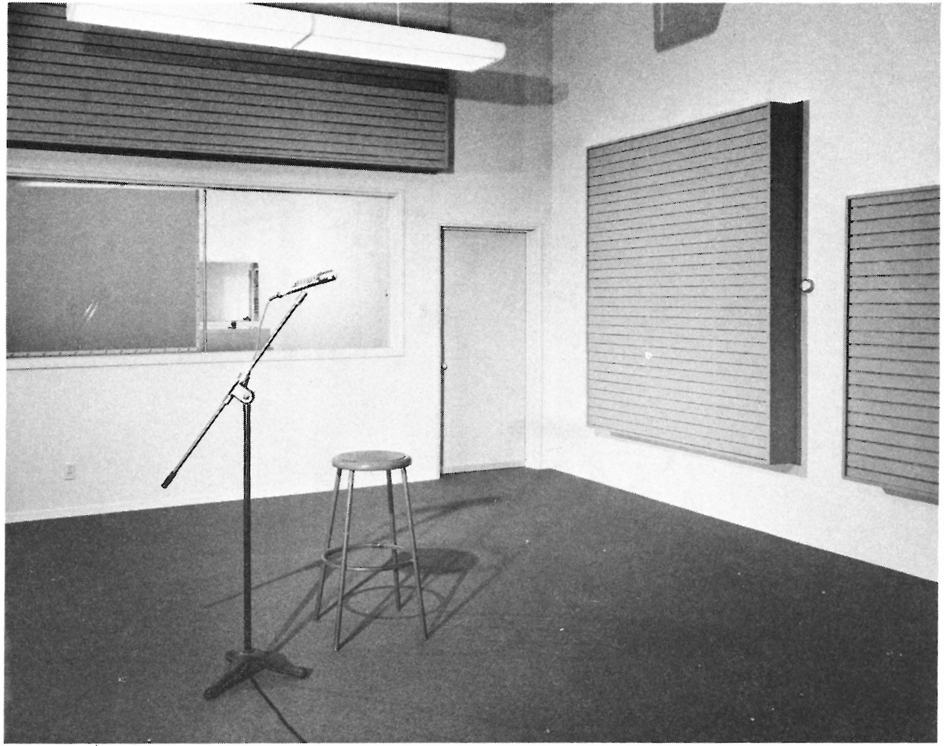


Fig. 16-15. A voice recording studio at Moody Institute of Science, Whittier, California. The slat resonators on the walls are designed for peak absorption at several low frequencies by various depth of the boxes.

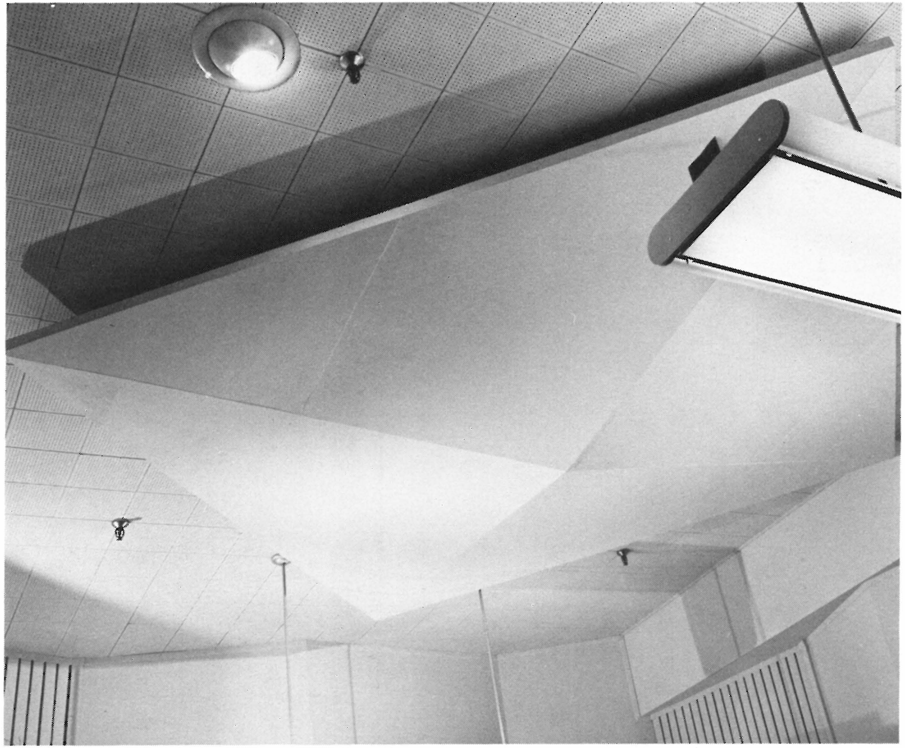


Fig. 16-16. The pyramidal ceiling feature contributes by (1) breaking up reflections between floor and ceiling, (2) reducing the effective area of already installed acoustical tile, (3) adding low frequency absorption by the panel effect, and (4) covering air inlet duct (the absorbent materials laid on its upper surface tending to absorb airborne noise coming through the duct).

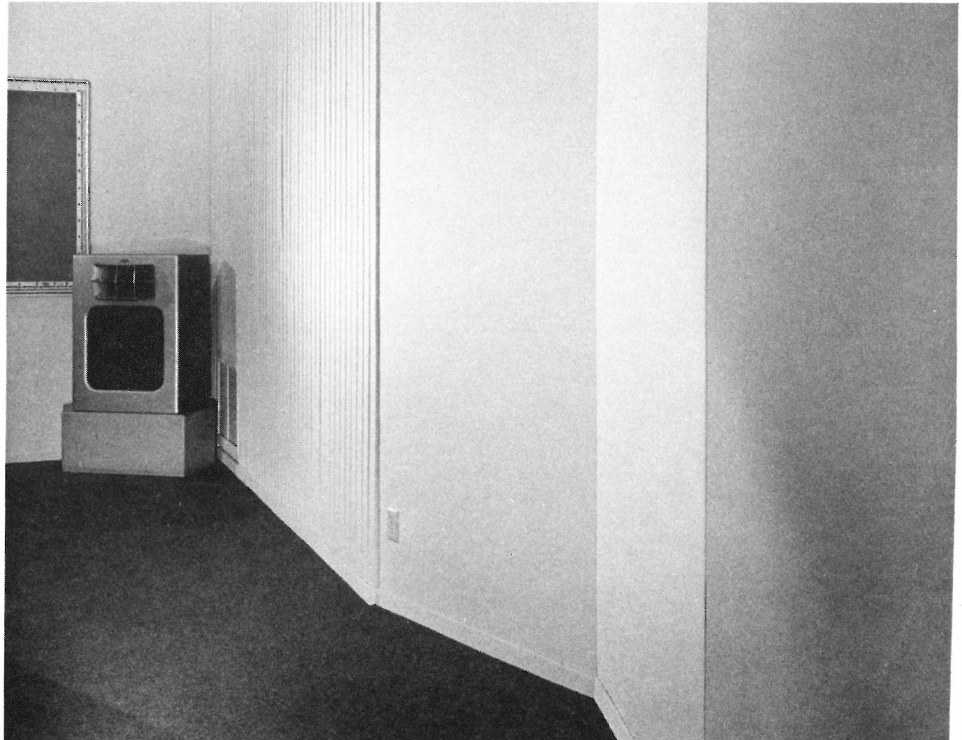


Fig. 16-17. Splayed panels are joined to slat absorbers. The wall opposite this one employs the same elements but reversed in position so that slats face splayed panels.

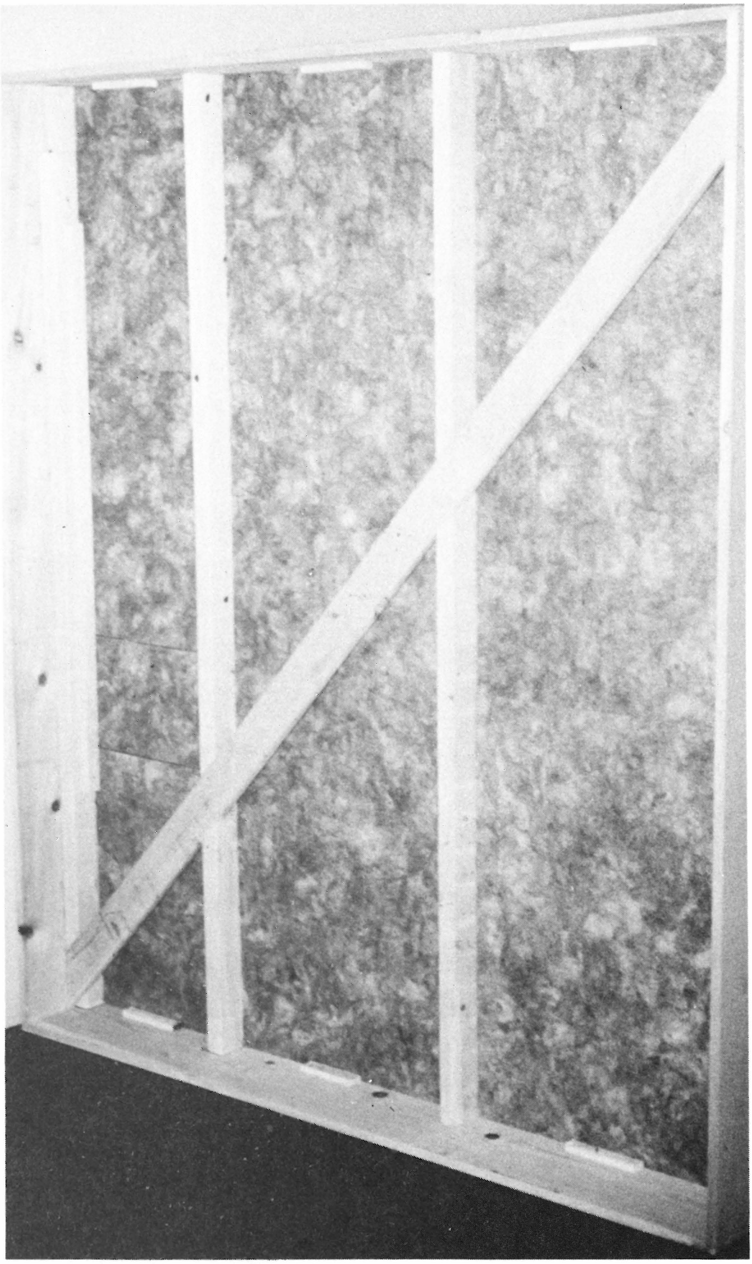


Fig. 16-18. Rear view of low frequency slat absorber such as those mounted on the wall in Fig. 16-15. The glass fiber board is semirigid, requiring little mechanical support.

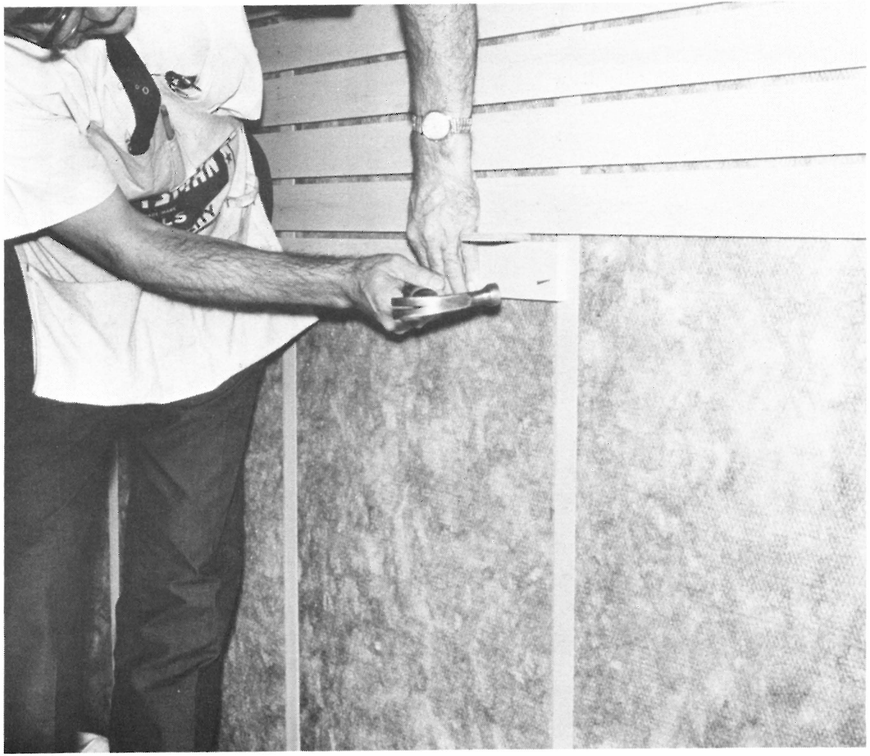


Fig. 16-19. The method of constructing the slat absorbers below the polys of Fig. 9-21. (Moody Institute of Science photos, Journal of the Audio Engineering Society.)

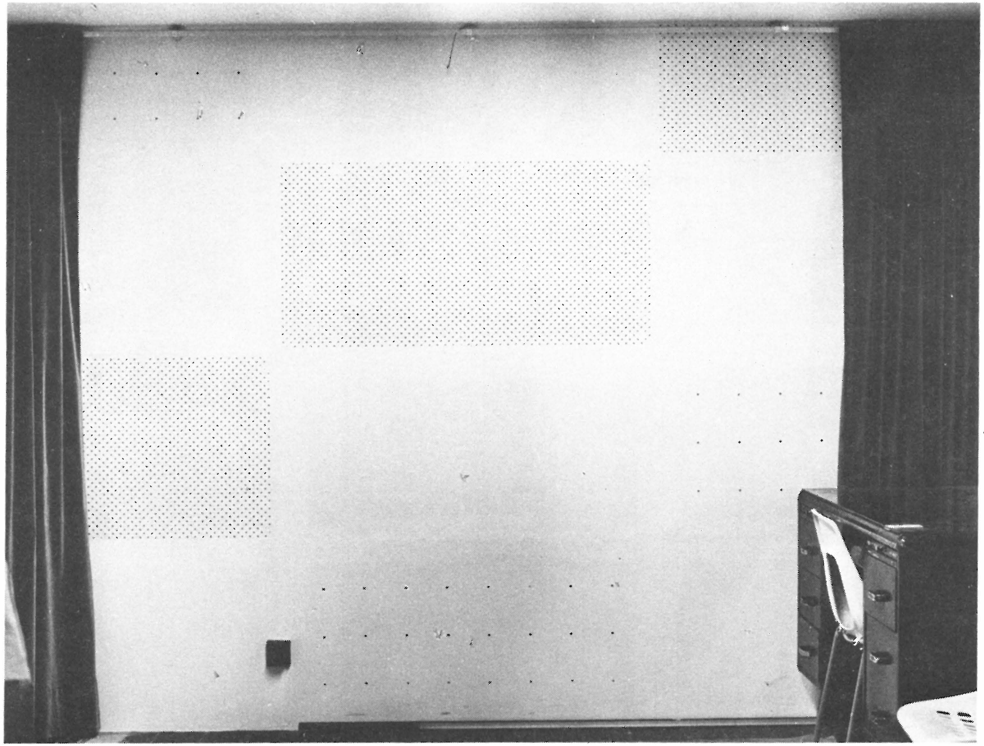


Fig. 16-20. The studio of the American Assemblies of God, located in Sha Tin, New Territories, Hong Kong, is used for recording radio programs. The dimensions are approximately 12 × 17 ft by 7′4″ ceiling height with the control room window set diagonally across one corner. This very small studio was designed by Mr. John L. Wheatley of the Far East Broadcasting Association of England. Mr. Wheatley has utilized the modular approach, basically, with 2×8″ lumber on edge forming the modules completely hidden by the covering materials on the ceiling and all walls. On this west wall a drapery can be drawn over the modules for limited control of reverberation.

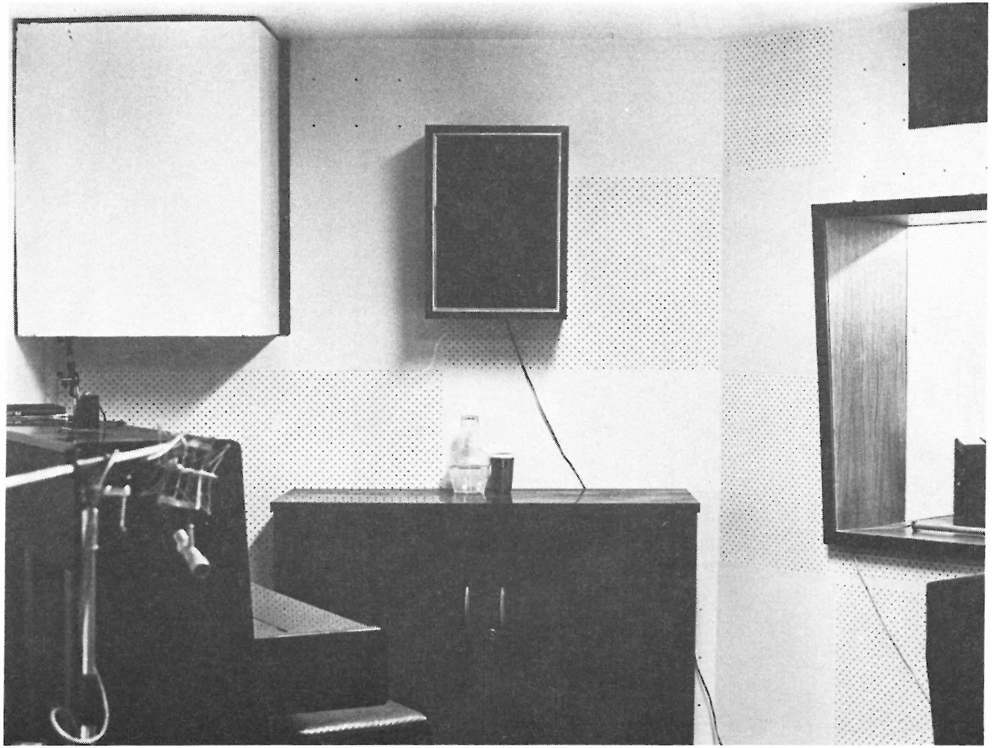


Fig. 16-21. East wall of the same studio showing the angled control room wall. The air conditioning duct is upper left. Three types of modules are employed with perforation percentages of 5.5%, 0.12%, and 0%.

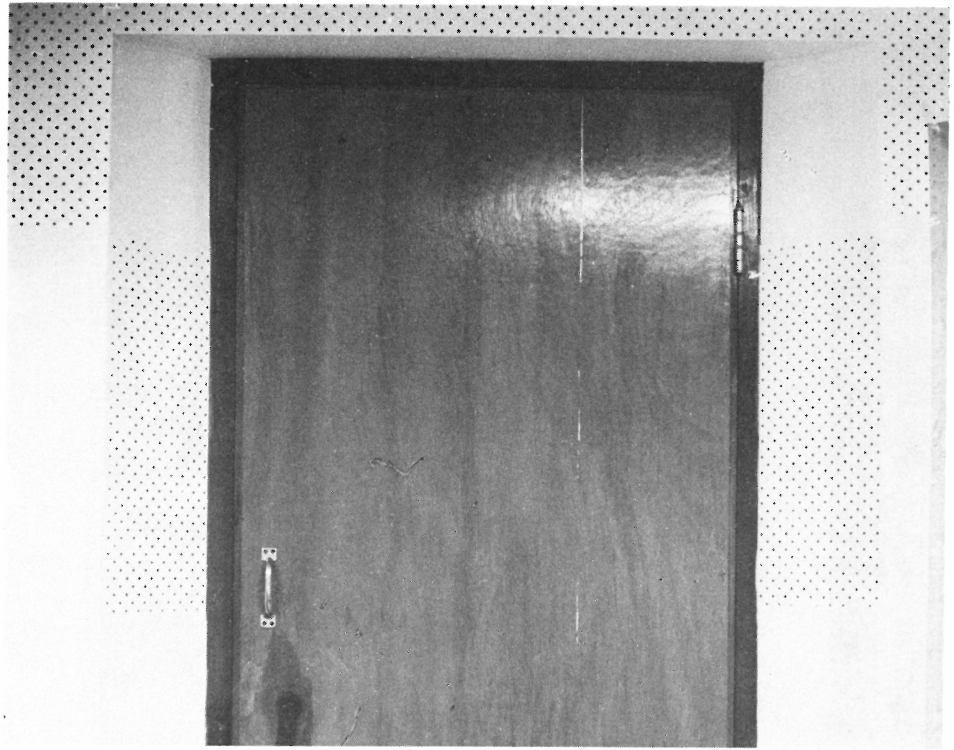


Fig. 16-22. The door is in the south wall. At a doorway, the built-out wall is apparent.

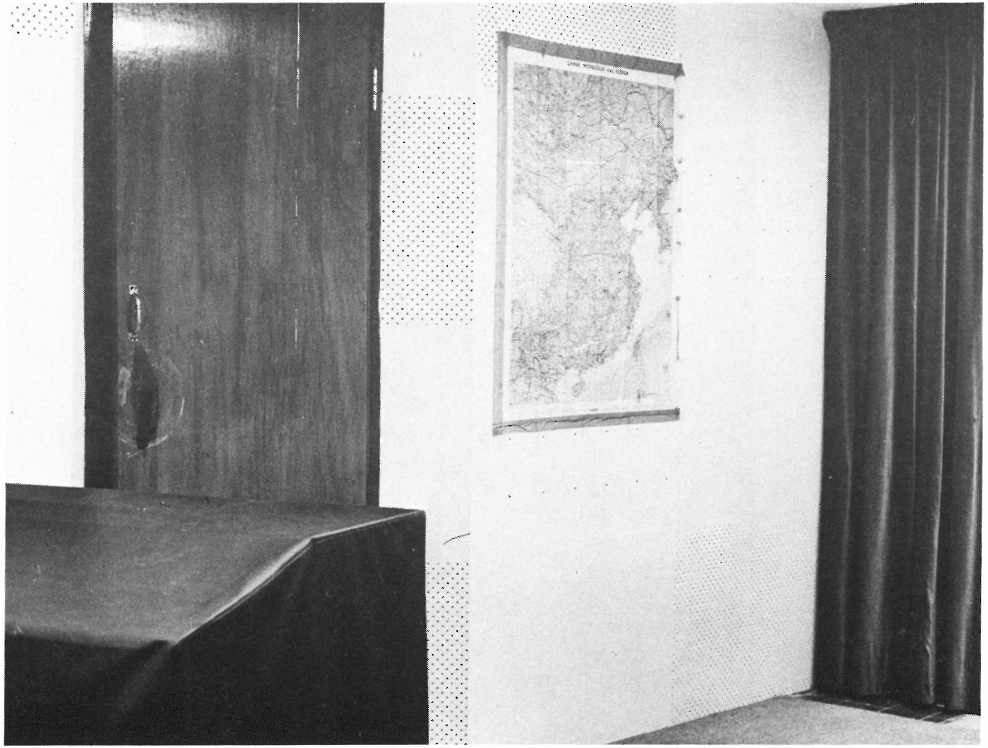


Fig. 16-23. An effort has been made to have modular sections of one type opposing sections of another type on the opposite wall. Estimations indicate that the reverberation time is close to the optimum value of 0.3 second for a volume of 1560 cu ft.

Fig. 16-24. A large poly dominates one end of the small Hong Kong Baptist College studio used for instruction in radio production. This poly contributes to low frequency absorption as well as the diffusion of sound in the studio.

Fig. 16-25. A view of the poly before the outer "skin" was applied. The frame and bulkheads were sealed to the plaster wall with a caulking compound, separating each compartment from its neighbors and from the studio space. Glass fiber batts are mounted in each compartment. The skin is ¼″ plywood to which ⅛″ finish veneer has been glued.

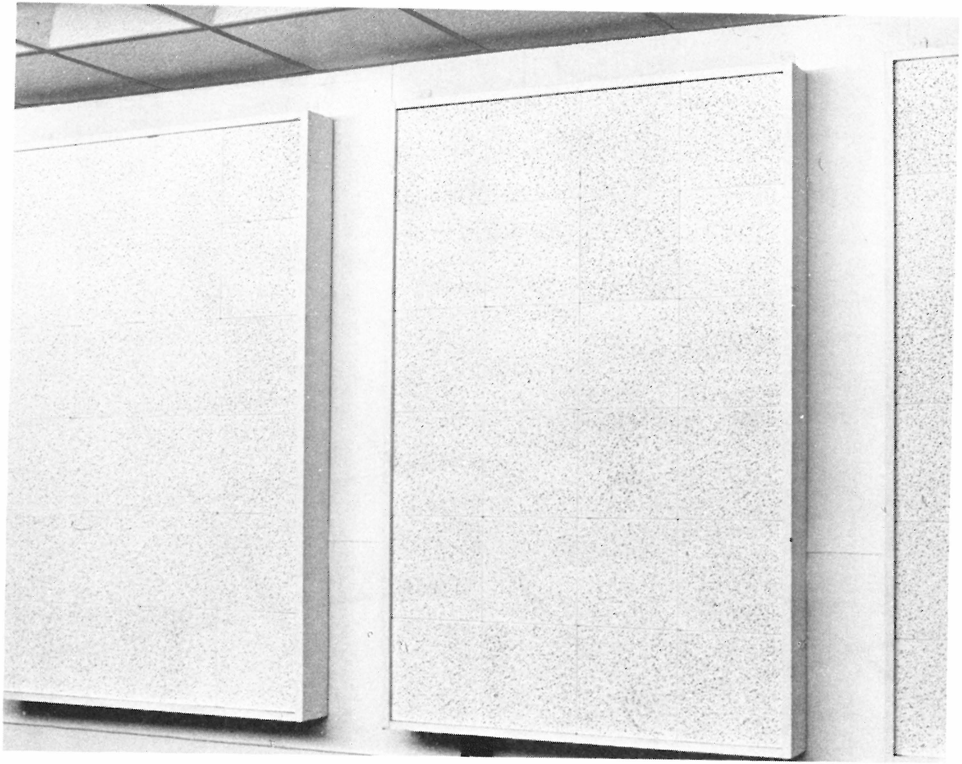


Fig. 16-26. Three reversible panels covered with Johns Manville ¾″ Tempertone tile on one side and ¾″ plywood on the other provide some control of reverberation.

Fig. 16-27. A suspended ceiling 19″ below the plaster ceiling provides good absorption through the band. This allows the use of practical vinyl tile on the floor without floor-ceiling reflection problems. The actual measured values of reverberation are:

Frequency (Hz)	T_{60} (seconds)
125	0.35
250	0.37
500	0.38
1000	0.35
2000	0.47
4000	0.44

These measurements reveal that a small amount of absorption is needed in the 2000 to 4000 Hz region.

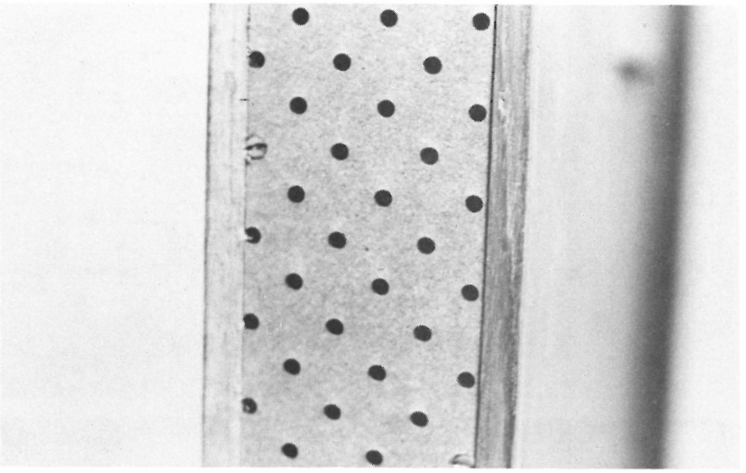


Fig. 16-28. Absorbent door edge to help in trapping outside noise passing through the inevitable crack between door and jamb. This door is constructed according to the sketches of Figs. 8-10 and 8-11.

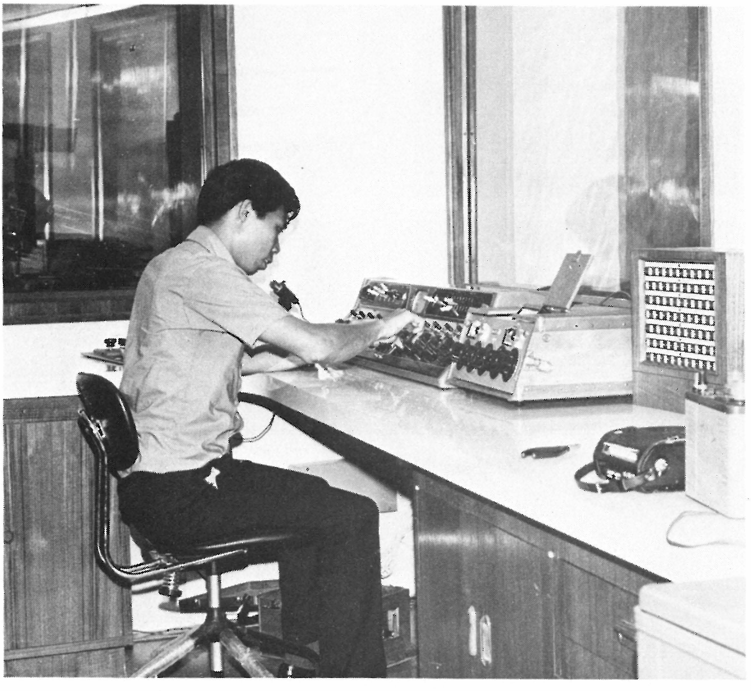


Fig. 16-29. The double glass window between control room and studio is patterned after the design of Fig. 8-9. (Hong Kong Baptist College photos.)

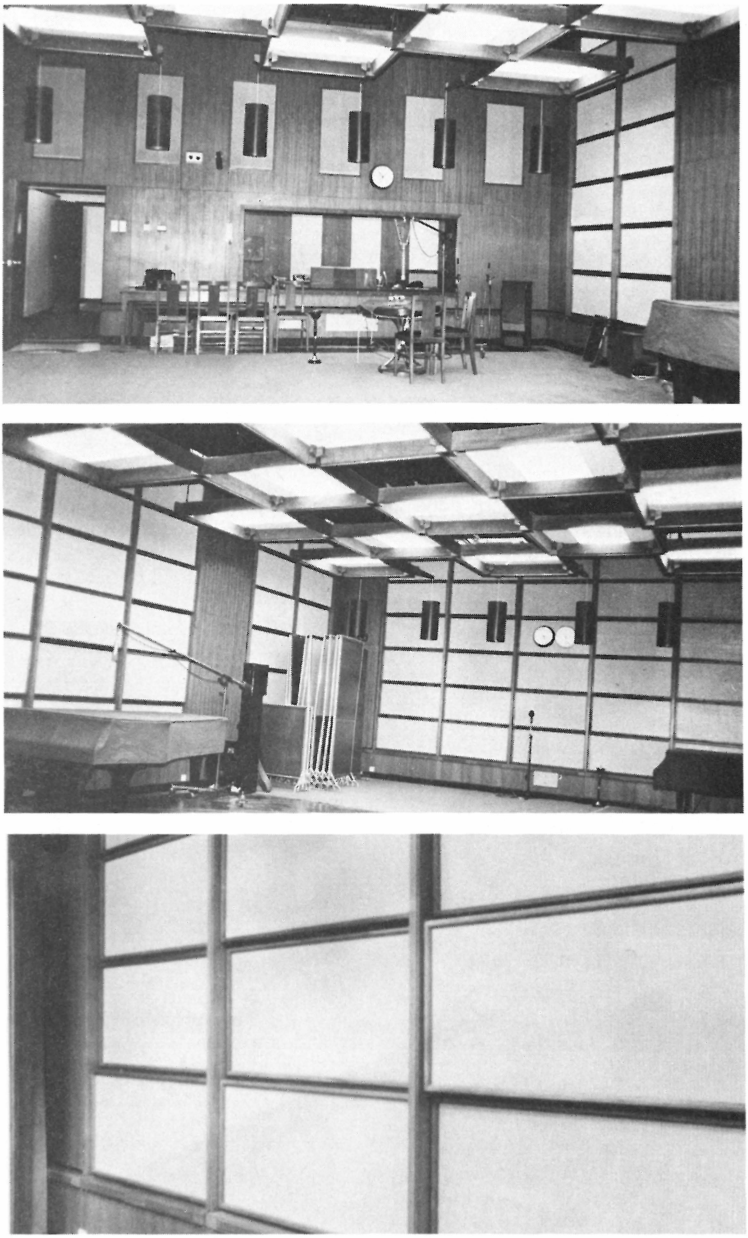
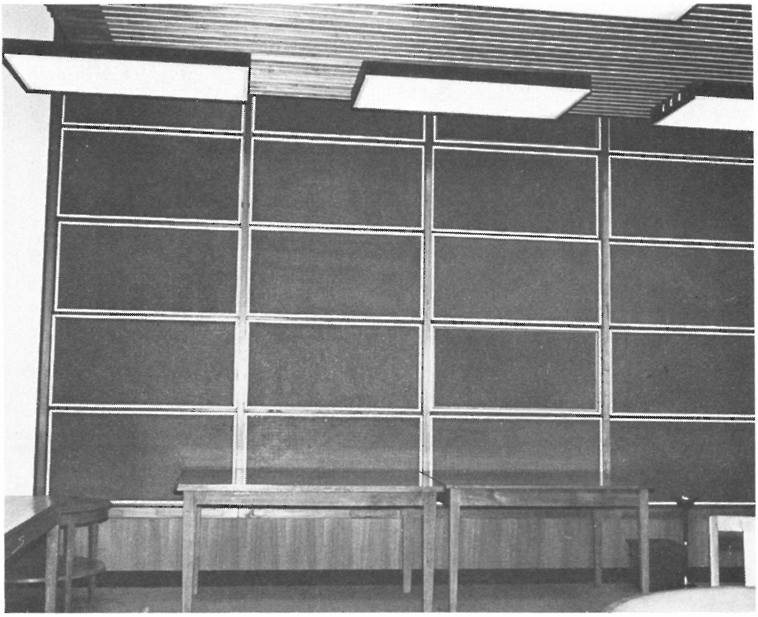


Fig. 16-30, 16-31, and 16-32. Views of the music studio at Radio Hong Kong. Modules after the general BBC plan are used in the acoustical treatment of this studio. Note the various depths of modules. It is understood that modules of three different absorption patterns are used.

Fig. 16-33 and 16-34. Views of a typical smaller general purpose studio at Radio Hong Kong. Here again modules are used effectively in achieving proper acoustical conditions. Radio Hong Kong Studios and treatment were designed by Mr. Ian Campbell, Lecturer in Architecture, University of Hong Kong.



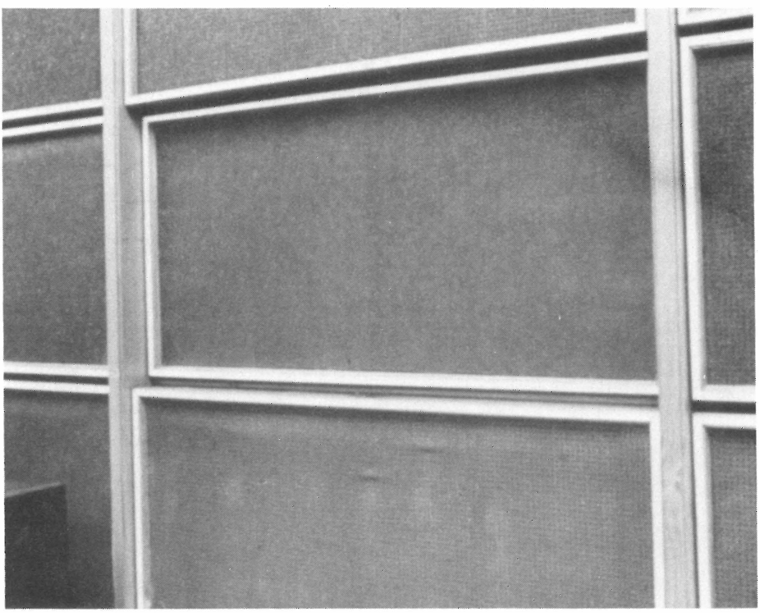


Fig. 16-35 and 16-36. Views of a typical smaller general purpose studio at Radio Hong Kong. Here again modules are used effectively in achieving proper acoustical conditions. Radio Hong Kong Studios and treatment were designed by Mr. Ian Campbell, Lecturer in Architecture, University of Hong Kong.

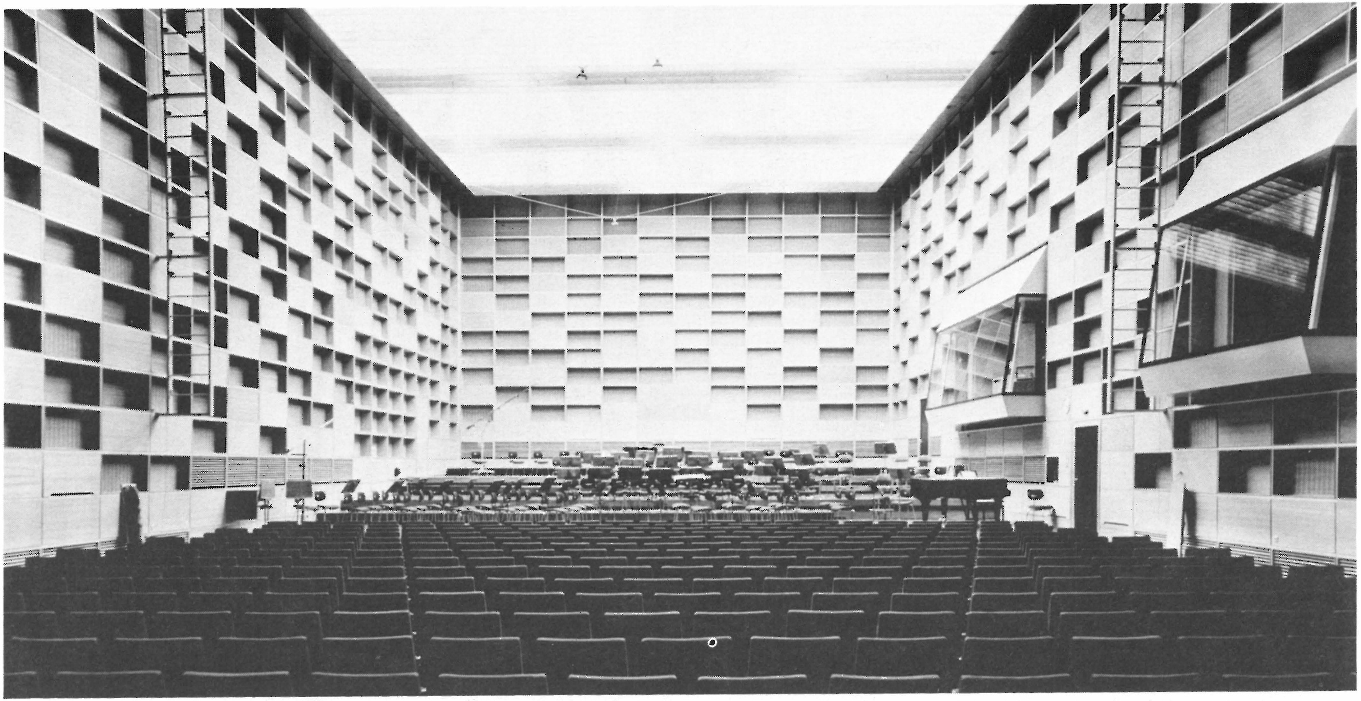


Fig. 16-37. Studio 1 at Bayerischer Rundfunk in Munich, West Germany. The very functional walls of this large studio are designed around acoustically absorbent modules. The distribution of these modules in a more or less random configuration aids diffusion of sound.

Fig. 16-38. The acoustical conditions of this room are dominated by shaped and spaced Hunter-Douglas aluminum panels backed by an acoustical pad. This arrangement yields excellent absorption in the 125-1000 Hz region.

Fig. 16-39. A view of another control room at Bayerischer Rundfunk. The studio in the distance is also treated acoustically by the module method. Note the perforated panel type of absorber to control resonances between the glass plates of the window. (Photos by FOTO-SESSNER courtesy Bayerischer Rundfunk.)

Fig. 16-40. Two views of the music studio of Voce Della Bibbia of Formigine, Italy. The standard modular wall absorbers (described in Fig. 16-41) are used in all three studios of this complex. Mounted on the upper surfaces of the suspended "clouds" are 16 perforated face absorbers 20×40 inches by 8 inches deep. The perforation percentage of the 3/16" cover is 0.34%. The measured reverberation time of this 6600 cu ft studio is:

Frequency (Hz)	T_{60} (seconds)
63	0.52
125	0.50
250	0.52
500	0.52
1000	0.57
2000	0.58
4000	0.54

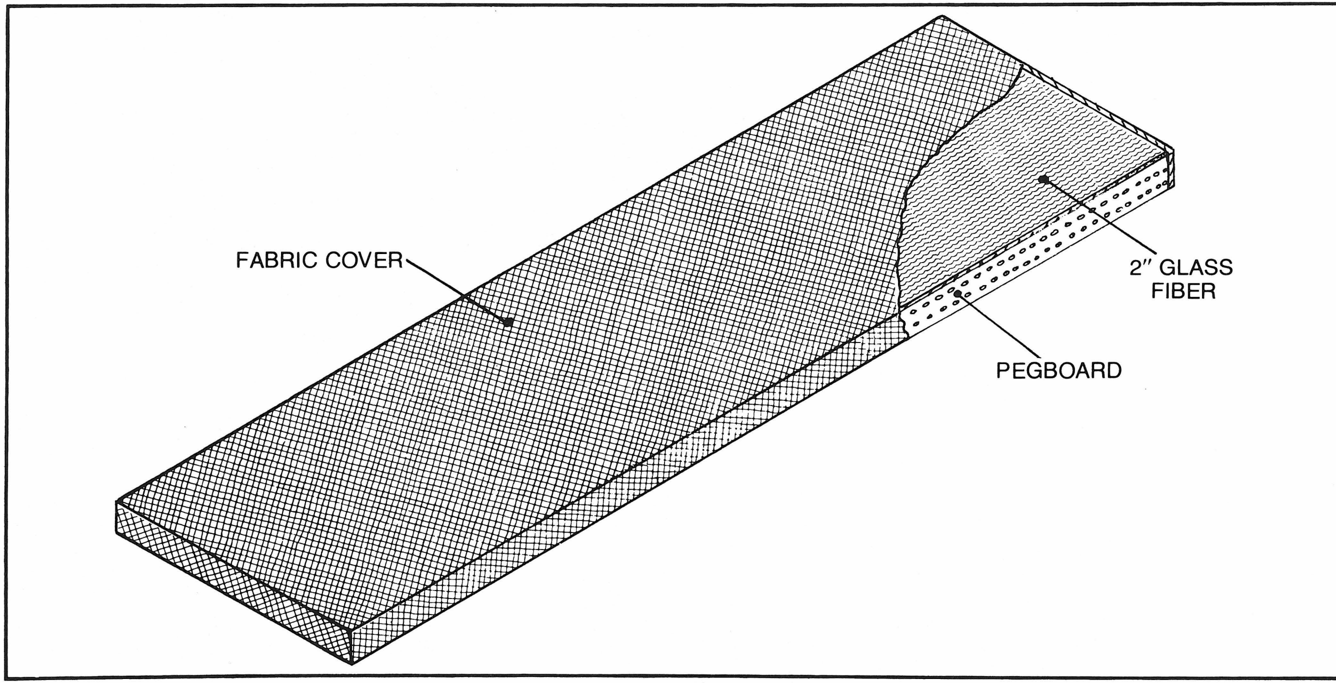


Fig. 16-41. The standard wall absorbing module used in all three studios of Voce Della Bibbia. These 2×6 ft panels have a backing of 3/16″ composition board to hold shape, edging of pegboard, a core of 2″ glass fiber of 3 lb/cu ft density, and a covering of fabric to "knock your eye out".

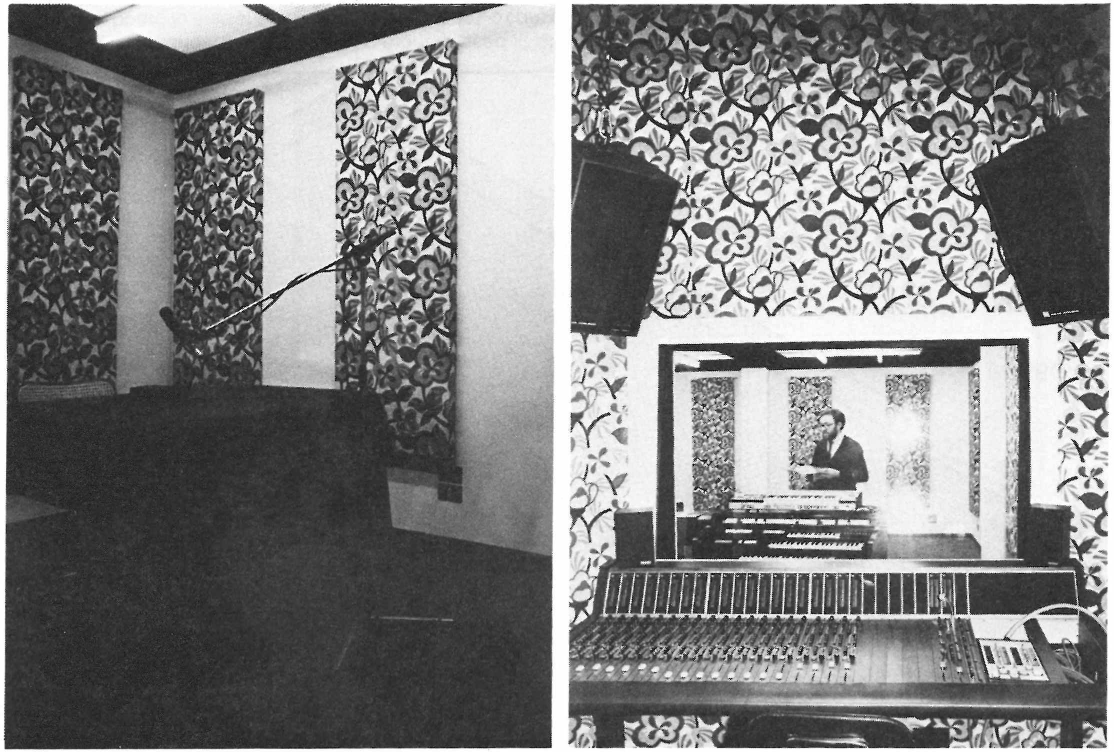


Fig. 16-42. Two views in one of the two voice studios of Voce Della Bibbia. These studios and their control rooms have minimum volumes of about 1500 cu ft. The studios and control rooms have identical dimensions, but the studios "stand on end" with the longest dimension vertically, while their control rooms have the longest dimension horizontally.

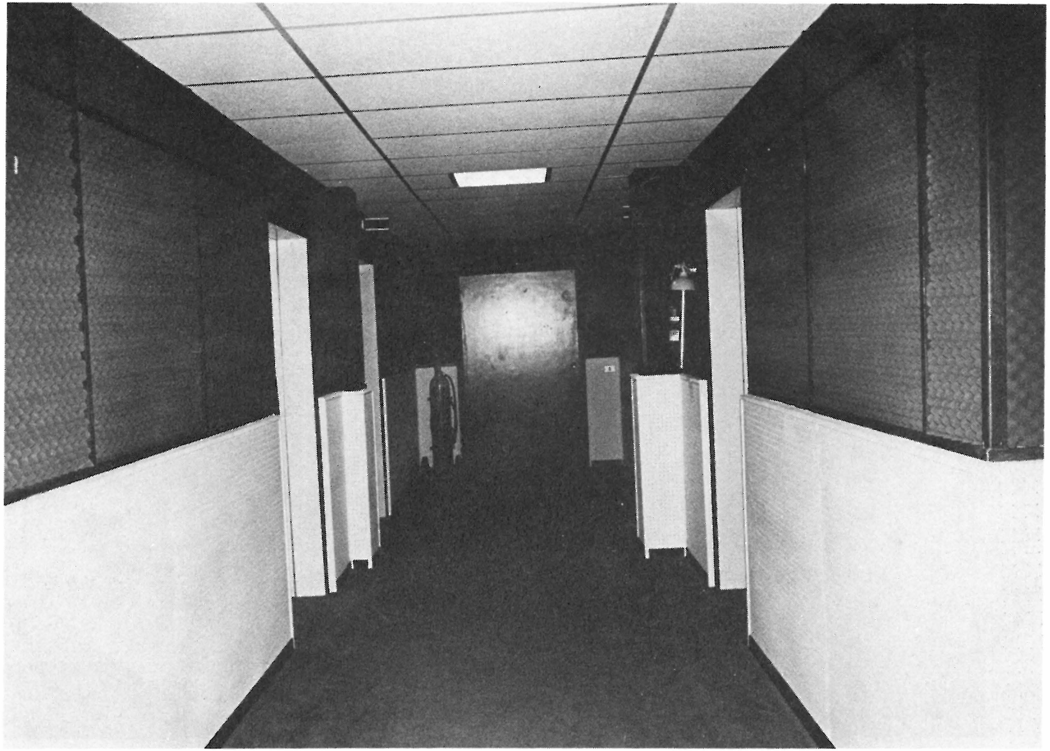


Fig. 16-43. Sound lock corridor at Voce Della Bibbia serving 3 control rooms, 3 studios, a tape duplication facility, and a tape library. Great care was exercised to make this corridor as absorbent as possible. The floor is heavily carpeted and the ceiling is of the conventional suspended type. The lower walls are of pegboard furred out 2″ with the void filled with glass fiber. The upper walls are covered with an "acoustically shaped" urethane open-cell foam.

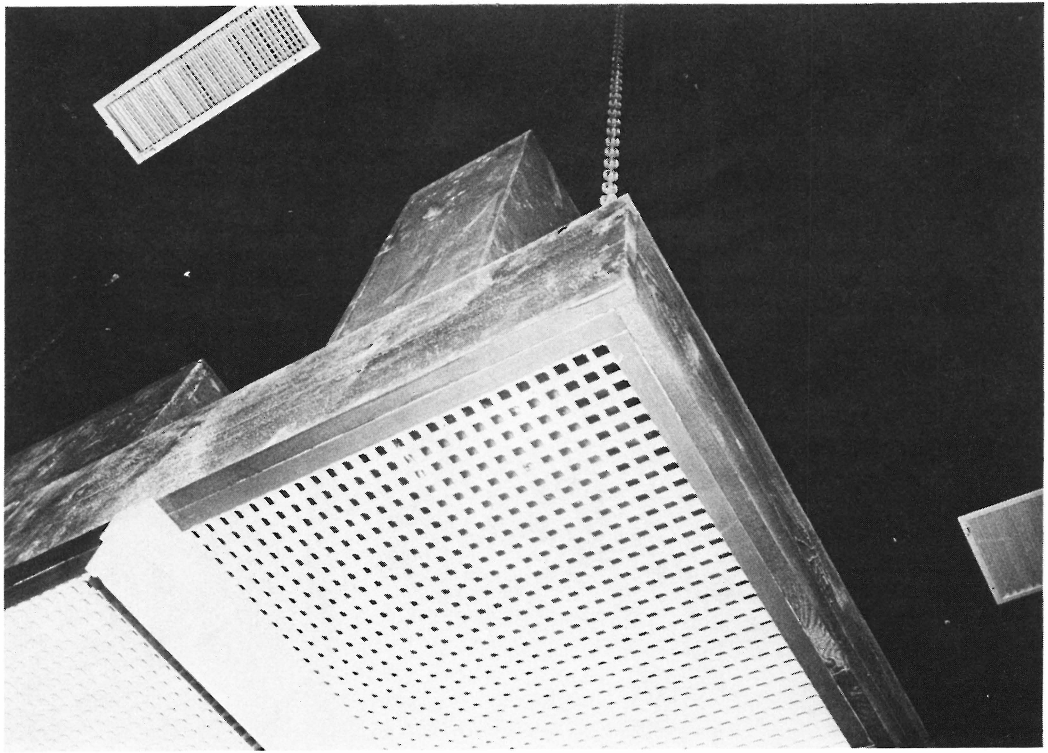


Fig. 16-44. Detail of the standard 6×8 ft suspended "cloud" used in all three studios of Voce Della Bibbia. A single unit is used in each of the voice studios and four in the music studio. Each "cloud" holds patches of glass fiber board (exposing both faces to the sound), perforated Helmholtz absorbers where required, and fluorescent lighting fixtures. These suspended elements also lower the visual ceiling for a more intimate effect and they also hide service runs. (Voce Della Bibbia photos arranged through the courtesy of Daniel Waldo of Back To The Bible Broadcast, Lincoln, Nebraska.)

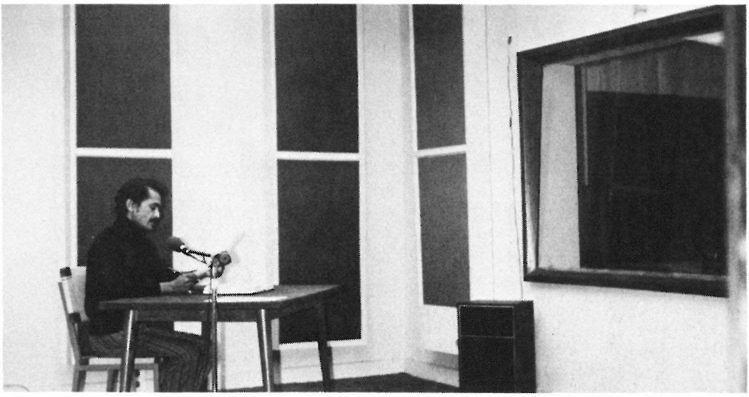


Fig. 16-45. The Speech-Drama Studio of the Good News Broadcasting Society of New Delhi, India. This complex of three studios and two control rooms occupies one end of the fourth floor of a high rise office building. Special provisions against structure borne noise include floating concrete floors, studio walls resting on the floating floors, and ceilings resiliently supported from the structure. Another studio of identical size accommodating straight voice recording is served by an auxiliary control room. All studios and control rooms except the music studio are the same size (2470 cu ft).

Fig. 16-46. Musicians in the music studio of the Good News Broadcasting Society of New Delhi. This studio has a volume of 3570 cu ft. Low frequency absorbers (2×4 ft by 8 inches deep) of the perforated face type are in the suspended "clouds".

Fig. 16-47. The Master Control Room of the Good News Broadcasting Society of New Delhi. This control room serves both the Speech-Drama Studio and the Music Studio. The 2×8 ft wall absorbers contain a 4″ thickness of 3 lb/cu ft density glass fiber. Because no carpets are used in the control rooms, the treatment of these spaces involves only 4″ thick glass fiber boards on walls and ceiling. (Good News Broadcasting Society photos arranged through the courtesy of Daniel Waldo of Back To The Bible Broadcast, Lincoln, Nebraska.)

Appendix
Selected Absorption Coefficients

Material POROUS TYPE	125 Hz	250 Hz	500 Hz	1 kHz	2 kHz	4 kHz	Reference (Chapter)
Drapes: cotton 14 oz/sq yd							
draped to 7/8 area	0.03	0.12	0.15	0.27	0.37	0.42	Mankovsky, ref 9-4
draped to 3/4 area	0.04	0.23	0.40	0.57	0.53	0.40	Mankovsky, ref 9-4
draped to 1/2 area	0.07	0.37	0.49	0.81	0.65	0.54	Mankovsky, ref 9-4
Drapes: medium velour, 14 oz/sq yd							
draped to 1/2 area	0.07	0.31	0.49	0.75	0.70	0.60	Mankovsky, ref 9-4
Drapes: heavy velour, 18 oz/sq yd							
draped to 1/2 area	0.14	0.35	0.55	0.72	0.70	0.65	Compendium, ref 9-1
Carpet: heavy on concrete	0.02	0.06	0.14	0.37	0.60	0.65	Compendium, ref 9-1
Carpet: heavy on 40 oz hairfelt	0.08	0.24	0.57	0.69	0.71	0.73	Compendium, ref 9-1

Material	125 Hz	250 Hz	500 Hz	1 kHz	2 kHz	4 kHz	Reference (Chapter)
POROUS TYPE							
Carpet: heavy with latex backing on foam or 40 oz hair-felt	0.08	0.27	0.39	0.34	0.48	0.63	Compendium, ref 9-1
Carpet: indoor/ outdoor	0.01	0.05	0.10	0.20	0.45	0.65	Seikman, ref 9-17
Acoustical tile, ave, 1/2″ thick	0.07	0.21	0.66	0.75	0.62	0.49	-----
Acoustical tile ave, 3/4″ thick	0.09	0.28	0.78	0.84	0.73	0.64	-----
MSC. BUILDING MATERIALS							
Concrete block, coarse	0.36	0.44	0.31	0.29	0.39	0.25	Compendium, ref 9-1
Concrete block, painted	0.10	0.05	0.06	0.07	0.09	0.08	Compendium, ref 9-1
Concrete floor	0.01	0.01	0.015	0.02	0.02	0.02	Compendium, ref 9-1
Floor: linoleum, asphalt tile, or cork tile on concrete	0.02	0.03	0.03	0.03	0.03	0.02	Compendium, ref 9-1
Floor: wood	0.15	0.11	0.10	0.07	0.06	0.07	Compendium, ref 9-1

Glass: large panes, heavy glass	0.18	0.06	0.04	0.03	0.02	0.02	Compendium, ref 9-1
Glass, ordinary window	0.35	0.25	0.18	0.12	0.07	0.04	Compendium, ref 9-1
<u>Drop Ceiling</u>							
Owens-Corning Frescor, painted, 5/8″ thick, Mounting 7	0.69	0.86	0.68	0.87	0.90	0.81	Compendium, ref 9-1
Plaster, gypsum or lime, smooth finish on tile or brick	0.013	0.015	0.02	0.03	0.04	0.05	Compendium, ref 9-1
Plaster: gypsum or lime, smooth finish on lath	0.14	0.10	0.06	0.05	0.04	0.03	Compendium, ref 9-1
Gypsum board: 1/2″ on 2×4s, 16″ on centers	0.29	0.10	0.05	0.04	0.07	0.09	Compendium, ref 9-1
<u>RESONANT ABSORBERS</u>							
Plywood panel: 3/8″ thick	0.28	0.22	0.17	0.09	0.10	0.11	Compendium, ref 9-1

Material	125 Hz	250 Hz	500 Hz	1 kHz	2 kHz	4 kHz	Reference (Chapter)
Polycylindrical:							
chord 45″ height 16″ empty	0.41	0.40	0.33	0.25	0.20	0.22	Mankovsky, ref 9-4
chord 35″ height 12″ empty	0.37	0.35	0.32	0.28	0.22	0.22	Mankovsky, ref 9-4
chord 28″ height 10″ empty	0.32	0.35	0.3	0.25	0.2	0.23	Mankovsky, ref 9-4
chord 28″ height 10″ filled	0.35	0.5	0.38	0.3	0.22	0.18	Mankovsky, ref 9-4
chord 20″ height 8″ empty	0.25	0.3	0.33	0.22	0.2	0.21	Mankovsky, ref 9-4
chord 20″ height 8″ filled	0.3	0.42	0.35	0.23	0.19	0.2	Mankovsky, ref 9-4
Perforated Panel 5/32″ thick 4″ depth 2″ glass fiber							
Perf: 0.18%	0.4	0.7	0.3	0.12	0.1	0.05	Mankovsky, ref 9-4
Perf: 0.79%	0.4	0.84	0.4	0.16	0.14	0.12	Mankovsky, ref 9-4

Perf: 1.4%	0.25	0.96	0.66	0.26	0.16	0.1	Mankovsky, ref 9-4
Perf: 8.7%	0.27	0.84	0.96	0.36	0.32	0.26	Mankovsky, ref 9-4
8″ depth 4″ glass fiber							
Perf: 0.18%	0.8	0.58	0.27	0.14	0.12	0.1	Mankovsky, ref 9-4
Perf: 0.79%	0.98	0.88	0.52	0.21	0.16	0.14	Mankovsky, ref 9-4
Perf: 1.4%	0.78	0.98	0.68	0.27	0.16	0.12	Mankovsky, ref 9-4
Perf: 8.7%	0.78	0.98	0.95	0.53	0.32	0.27	Mankovsky, ref 9-4
With 7″ air space plus 1″ mineral fiber of 9-10 16 cu ft/ lb density, 1/4″ cover							
Wideband, 25% perf or more	0.67	1.09	0.98	0.93	0.98	0.96	BBC, ref 9-18
Midpeak, 5% perf	0.60	0.98	0.82	0.90	0.49	0.30	BBC, ref 9-18
Lo-peak, 0.5% perf	0.74	0.53	0.40	0.30	0.14	0.16	BBC, ref 9-18
With 2″ air space filled with mineral fiber, 9-10 lb/ cu ft density							
Perf: 0.5%	0.48	0.78	0.60	0.38	0.32	0.16	BBC, ref 9-18

Index

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